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No. 24—EXAMPLES OF CALCULATING DESIGNS

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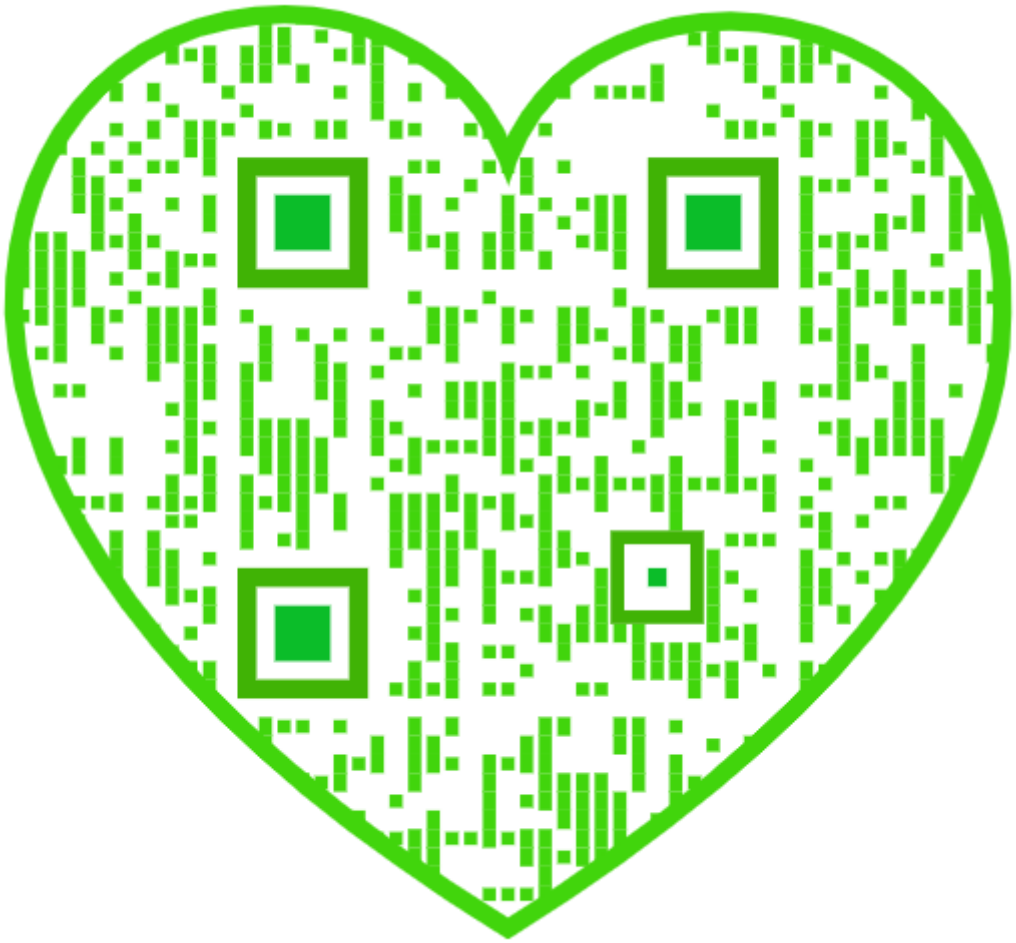

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CHAPTER I.

CHARTS IN DESIGNING.

Charts which are used for recording data or for assisting the designer in proportioning his work or in making calculations, may be classified, according to the purpose for which they are intended, under two general heads, *i. e.*, Record Charts, and Calculating Charts.

To the first class belong cards from all recording devices, and diagrams of any kind laid out from known values, such as graphical representations of results of tests. Such charts may constitute a record pure and simple; they may be for the purpose of better showing the existing relations of the quantities involved, for ascertaining mean, intermediate, or proportional values, or they may be developed with a view to discovering losses, irregularities, errors, etc.

Record Charts.

The following development of one of a series of tests on the power required to drive rivets is a fair illustration of the applicability of record charts:

Experiment on ¾-inch rivets, holes punched 13-16 inch at top, about ⅞ inch at bottom, 1.4 inch grip, about 1½ inch projection of blank before driving. The rivets were driven on an Allen riveter, the frame having been previously calibrated as a spring balance by measuring the deflection in thousandths of an inch with a micrometer, for known loads suspended from the die. Chart Fig. 1 shows the calibration of the frame.

In Chart Fig. 2 the full line represents graphically the results of the experiments as given in Table I. The dotted curve represents allowances for efficiency and inequalities in length of the rivets, plotted with a view of determining the energy required for a power driven machine. In this test the mean average pressure during the period of driving was 40,925 pounds. The distance passed over under this mean pressure was 29-64 inch.

$$\frac{40{,}925 \times 29}{64 \times 12} = 1{,}545 \text{ foot-pounds of energy.}$$

$$\text{With allowances as per dotted line} = \frac{38{,}000 \times 1.25}{12} = 3{,}960 \text{ foot-pounds.}$$

$$\text{If exerted every 3 seconds the H. P. to drive} = \frac{3{,}960 \times 60}{33{,}000 \times 3} = 2.4.$$

Of course allowances would have to be made for other losses and for extra stroke.

The uses which have developed for charts have become as numerous as the different branches of industry with which they are associated.

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Their construction, when taken up in detail, is even more varied than their uses. Cold figures never convey to the mind as clear a conception of relative values as a graphical representation, and in the case of experiments the tabulated data often brings to view points when charted, which would not otherwise be noted.

Frequently a record chart is used to interpolate between known values, obtaining unknown ones, as was the calibration of the riveter frame in the foregoing example.

Similar to the last mentioned class of charts, but closely allied to Calculating Charts, belong stress diagrams, and all kindred matter in the art of graphostatics, so much used in the determination of stresses in framed structures, bending moments in beams, etc.

Classes of Calculating Charts.

Calculating Charts may be defined as those which express mathematical relations of quantities. They are of two varieties: Those

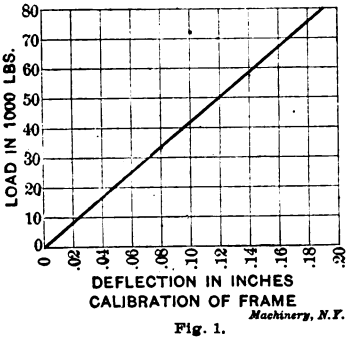


Fig. 1.

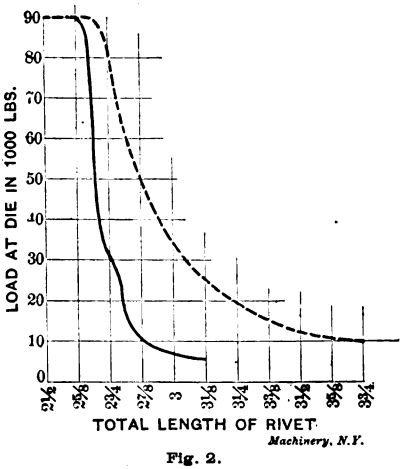


Fig. 2.

upon which certain mathematical operations may be performed according to any sequence, the quantities being either concrete or abstract and of any denomination and measure desired; and those designated for dealing only with concrete numbers of definite measure and denomination, the quantities involved having a fixed mathematical relation and sequence.

Slide Rules.

Under the first subdivision, if this liberty of classification be permissible, come all forms of calculating instruments, such as the slide rule, Sexton's omnimeter, Thatcher's calculating machine, etc. The principles upon which these operate are quite generally known, leaving little to be said in this connection. There is, however, a spirit of distrust far too prevalent regarding the accuracy of calculations performed upon a slide rule. Because one reads from the rule 975 pounds

as the weight of a casting and the actual multiplications give 974.235 pounds, is that any reason why the rule should not be used for estimating weights? In all probability one would set the weight down as 980 pounds for sake of even figures, and if a casting, it might come out of the sand weighing over 1,000 pounds.

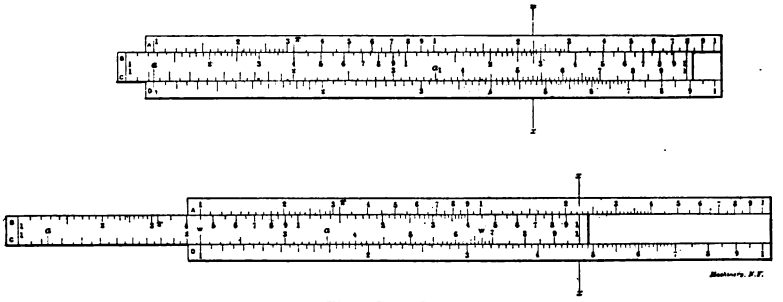
In calculating stresses, for instance, who cares for a hundred pounds

TABLE I

No. of Rivets.	Deflection of Frame.	Pressure at Die, pounds.	Total Length of Rivet Head to Head	Height of Head being Formed.	Remarks.
1	.016	6000	$3\frac{7}{64}$	$\frac{9}{32}$	End stove up.
2	.022	9000	$2\frac{59}{64}$	$\frac{17}{64}$	End stove up $\frac{1}{8}$" more.
3	.036	15000	$2\frac{57}{64}$	$\frac{1}{4}$	Head commencing to form in die.
4	.063	26000	$2\frac{55}{64}$	$\frac{15}{64}$	Head formed in part only.
5	.095	39000	$2\frac{53}{64}$	$\frac{7}{32}$	Head not completely formed.
6	.150	62000	$2\frac{51}{64}$	$\frac{13}{64}$	Fairly well formed head.
7	.208	82600	$2\frac{49}{64}$	$\frac{3}{16}$	Better head, hole well filled.
8	.214	87800	$2\frac{47}{64}$	$\frac{11}{64}$	Some hotter than last rivet; good head.

more or less? If the assumed fiber stress be 10,000 pounds per square inch, an error of even 500 pounds is only 5 per cent. The rule has a wide sphere of usefulness, and those persons who cry out so against it as an unreliable instrument are only lacking in judgment as to where and where not extreme accuracy is required.

The settings, Figs. 3 and 4, illustrate the location of two much used



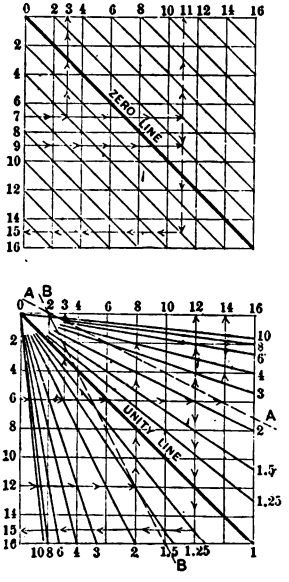
Figs. 3 and 4.

constants which may be applied to any slide rule, and after used a short time their utility will become apparent. In Fig. 3, set 4 on the B scale under π on the A scale; at left index make mark a on the C scale. Set second 4 under π and mark a_1. Now, setting a or a_1 to any diameter on the D scale gives the area on the A scale opposite either index; then, setting the runner to a given length of cylinder on B gives the volume on A.

Set the rule as shown in Fig. 3, with the runner at 283 on the *B* scale as indicated at *x*; set the right index of the slide at this line *xx* and at the left index mark *W*, Fig. 4, on the *C* scale. Mark second *W* in line with the middle 1 of the *A* scale. Setting either *W* to any diameter on the *D* scale gives the weight per inch of round steel bars on the *A* scale; setting the runner to the length of the bar on *B* gives the weight of the bar. As an aid to the memory for the uses of these points, *a* stands for areas and *W* for weights.

Calculating Charts.

Taking up Calculating Charts in detail, simple mathematics may be divided into three elementary operations, each class consisting of a



Figs. 5 and 6.

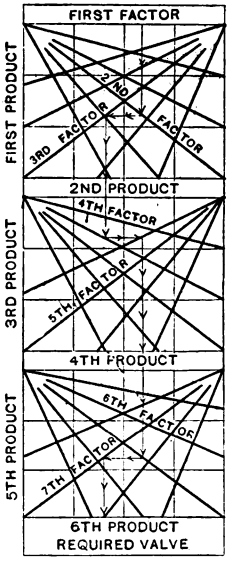


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Fig. 7.

couplet in which the one member is the inverse of the other, *viz.*: addition and subtraction; multiplication and division; raising to a power or extracting a root.

Chart Fig. 5 illustrates how the operations of addition and subtraction may be performed graphically.* Following the fine line in the direction indicated by the arrow heads represents operations as follows: $7 - 4 = 3$, $7 + 4 = 11$, $7 + 4 + 4 = 15$, $9 + 2 = 11$, $9 + 2 + 4 = 15$. Following these lines in the opposite direction would perform the inverse operation. It will be noticed that when the zero line is crossed to find the next value, addition is performed, but when the zero line is not so crossed subtraction results, *i. e.*, the quantities

* See also MACHINERY'S Reference Series No. 19, Use of Formulas in Mechanics, page 19: Graphical Methods of Solving Problems.

represented by the diagonal lines on the same side of the zero line as the quantity it is desired to combine with, are negative.

Chart Fig. 6 illustrates in a similar manner the operations of multiplication and division. The arrows indicate the following operations: $6 \div 2 = 3$, $6 \times 2 = 12$, $6 \times 2 \times 1.25 = 15$. The line *A-A* represents multiplication by 2 combined with addition of a constant whose value is also 2, thus: $6 \times 2 + 2 = 14$. This line is drawn parallel to the multiplier (in this case parallel to 0-2) and has its point of origin at the constant. Line *B-B* represents division by 2 with addition of a constant, thus: $12 \div 2 + 2 = 8$. It will be noticed that in addition and subtraction the parallel diagonal lines have their origin at the value they represent, and their terminus at a similar value on the side adjacent, if the scale on that side were reversed; while in multiplication and division the lines are divergent, and have their origin at the common zero point of the two scales.

In Figs. 5 and 6 the operations may be carried on continuously within the range of their readings, thus: $[(9 + 2 = 11) + 4 = 15] - 2 = 13$, and similarly, $[(6 \times 2 = 12) \div 1.5 = 8] \times 1.25 = 10$. This is dealing with abstract numbers, however. A single line chart handling concrete values will combine two quantities and give the result of whatever the operation may be. A chart having two sets of sloping lines will combine three values giving the result. If more than three values are to be combined, a multiple chart, as illustrated in Fig. 7, may be used. The following is a concrete example of the last-mentioned chart, illustrating plain multiplication and division. (See Fig. 8.)

Tank Chart.

Required: the thickness of tanks of various diameters and heights for containing different fluids. The formula for thin cylinders is as follows: $t = \frac{pd}{2sy}$, where t = thickness in inches, p = pressure in pounds per square inch, d = diameter in inches, s = allowable working stress in pounds per square inch, and y = the efficiency of the riveted joint.

Let D = diameter in feet, H = head in feet, and G = specific gravity of fluid; then

$$\frac{0.433\,HG\,12\,D}{2\,sy} = 2.598\,\frac{HGD}{sy}$$

In operations of multiplication and division the order in which the factors are taken does not affect the value of the result; it is therefore advisable to choose those quantities which are most constant for any given problem as the first factors of the chart. In this case the order has been chosen as follows: diameter, specific gravity, stress, efficiency of joint, and head.

Next in order of importance comes range of readings and scales. It is always desirable to make the range of a chart as great as is consistent with reasonable accuracy. In the case under consideration the

range of diameters has been chosen between 6 and 120 feet, but in order to secure this range three different scales have been used; 2 feet per space from 6 feet up to and including 20 feet; 5 feet per space from 20 feet up to and including 60 feet; and 10 feet per space from 60 feet

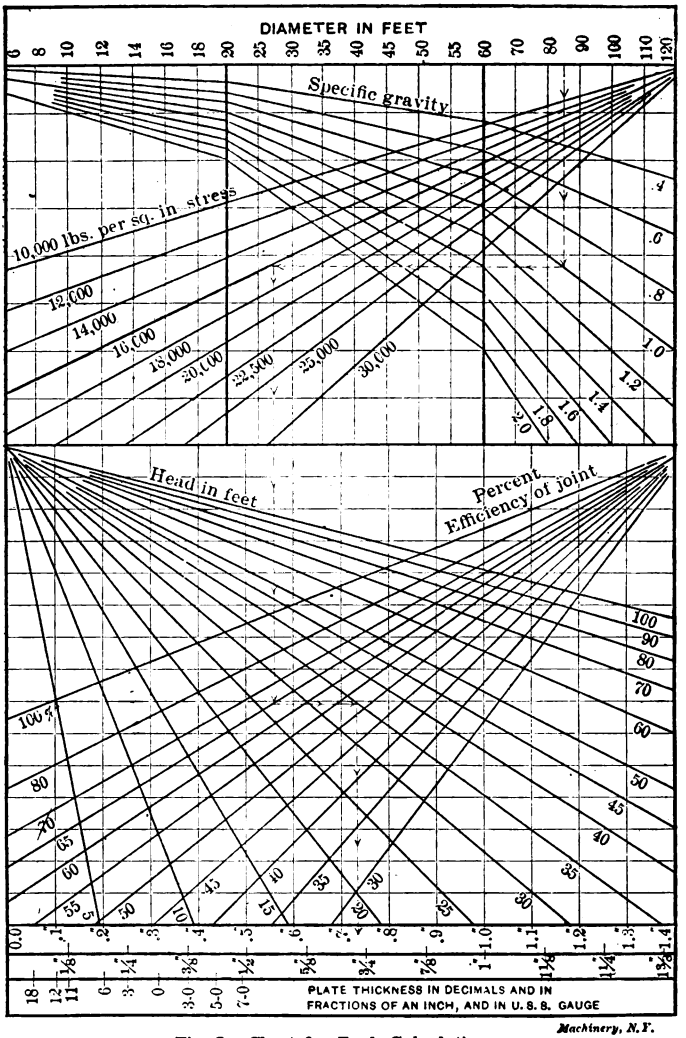

Fig. 8. Chart for Tank Calculations.

up to and including 120 feet. (See Fig. 8.) This change of scales necessitates three different points of origin for the lines representing the second factor, but the error of interpolation is thus kept within reasonable limits without making a chart of abnormal proportions.

To aid one in deciding upon appropriate scales and limits of read-

ings for the other factors and products, it is well to tabulate these factors and their resulting products, first choosing assumed limits of reading for the intermediate quantities which will produce the mini mum final value, then vice versa, and lastly, what may be considered a practical problem of maximum sizes. (See Table II.) From these values the range of the different products has been chosen, as indicated in the last two lines of the table. Their scale will be determined

TABLE II. TABULATED FACTORS AND PRODUCTS FOR USE IN SELECTING LIMITS OF READINGS AND SCALES.

Assumed Values.	First Factor, Diameter in Feet = D	Second Factor, Specific Gravity = G	First Product = $D\,G$	Third Factor, 1 ÷ stress = $\frac{1}{S}$	Second Product = $\frac{D\,G}{S}$	Fourth Factor, 1 ÷ Efficiency of Joint = $\frac{1}{y}$	Third Product = $\frac{D\,G}{Sy}$	Fifth Factor, Head in Feet × Constant = $H \times 2.598$	Fourth Product, Thickness in Inches = $2.598\,\frac{H\,G\,D}{Sy}$
Assumed values giving minimum final reading	6	0.65	3.9	$\frac{1}{25000}$	0.00015	$\frac{1}{0.80}$	0.00019	10×2.598	0.005
Assumed values giving maximum final reading	120	2.0	240	$\frac{1}{8000}$	0.03	$\frac{1}{0.50}$	0.06	100×2.598	15.6
Practical problem of maximum size	120	1.1	132	$\frac{1}{16000}$	0.00825	$\frac{1}{0.60}$	0.01375	40×2.598	1.43
Fixed upon as minimum reading	6	0.4	0	$\frac{1}{30000}$	0.0002	$\frac{1}{1.0}$	0.0	5×2.598	0.0
Fixed upon as maximum reading	120	2.0	160	$\frac{1}{10000}$	0.0086	$\frac{1}{0.30}$	0.015	100×2.598	1 4

by the size of the sheet it is desired to use. Since it is not necessary to have the points of origin lie within the border lines, it is desirable to keep the minimum readings greater than zero when this can be done without impairing the utility of the chart.

The problem illustrated by the arrows in Fig. 8 is a tank 85 feet in diameter, specific gravity = 1.0, 16,000 pounds fiber stress, 65 per cent efficiency of joint, and 35 feet head. The thickness is read to the nearest even fraction on the side of safety as ¾ inch.

Cone Pulleys.

The chart Fig. 10, for cone pulleys, illustrates the variable scale principle combined with subtraction and division. For crossed belts the belt will be of uniform tension when $A + A_1 = B + B_1 = C + C_1$, etc., Fig. 9, but for open belts the middle steps must be larger in diameter than for crossed belts.

The formula used for this case is as follows: (See Fig. 9.)

$$x = \frac{(A - A_1)^2 - (B - B_1)^2}{2\pi L}; \quad x_1 = \frac{(A - A_1)^2 - (C - C_1)^2}{2\pi L};$$

$$x_2 = \frac{(A - A_1)^2 - (D - D_1)^2}{2\pi L}, \text{ etc.}$$

where x = total correction to be added to the sum of $B + B_1$, x_1 = total correction to be added to the sum of $C + C_1$, etc., to preserve a uniform belt tension, the diameters used in the formula being to the center of the belt.*

Proceeding to construct a chart which will give the values of x, x_1, x_2, etc., for any case within its range, we first inspect the formula.

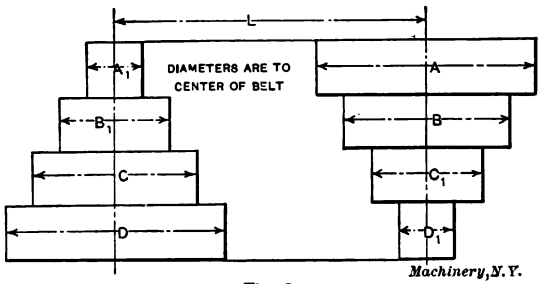

Fig. 9.

We have three main values involved $(A—A_1)^2$, $(B—B_1)^2$ and $2\pi L$, the difference between the first two being divided by the last. As 2π is a constant it does not enter as one of the variable factors. The first two factors are to be squared. This is best accomplished in the following manner: Lay out at the top of the chart in lead pencil a uniform scale (in this case the range has been chosen from 0 to 1,000); this represents the values of $(A—A_1)^2$; now spot off underneath this scale by the aid of a table the square roots of such numbers as will make a fairly uniform looking scale, inserting their values in ink, and erasing the former scale. In this case we wish to subtract from the first quantity $(A—A_1)^2$ a similar quantity $(B—B_1)^2$. Since the range of $(B—B_1)^2$ should be nearly the same as for $(A—A_1)^2$, lay off on one side of the chart the same uniform scale from 0 to 1,000, starting with 0 at the right-hand corner on the top scale. Now, parallel lines drawn from one value on the top scale to an equal value on the side scale will represent the values $(B—B_1)^2$. The lines have the same

* See MACHINERY'S Reference Series No. 14, Details of Machine Tool Design, page 3: Elementary Principles of Cone Pulleys and Belts.

values as their point of origin on the top scale and need not be numbered. The squaring operation has already been performed by the variable scale. The values on the side now represent the result of the first operation and are equal to $(A—A_1)^2 — (B—B_1)^2$, which value is to be divided by $2\pi L = 6.283L$.

Now lay off a uniform scale on the bottom for values of x, x_1, x_2, etc.,

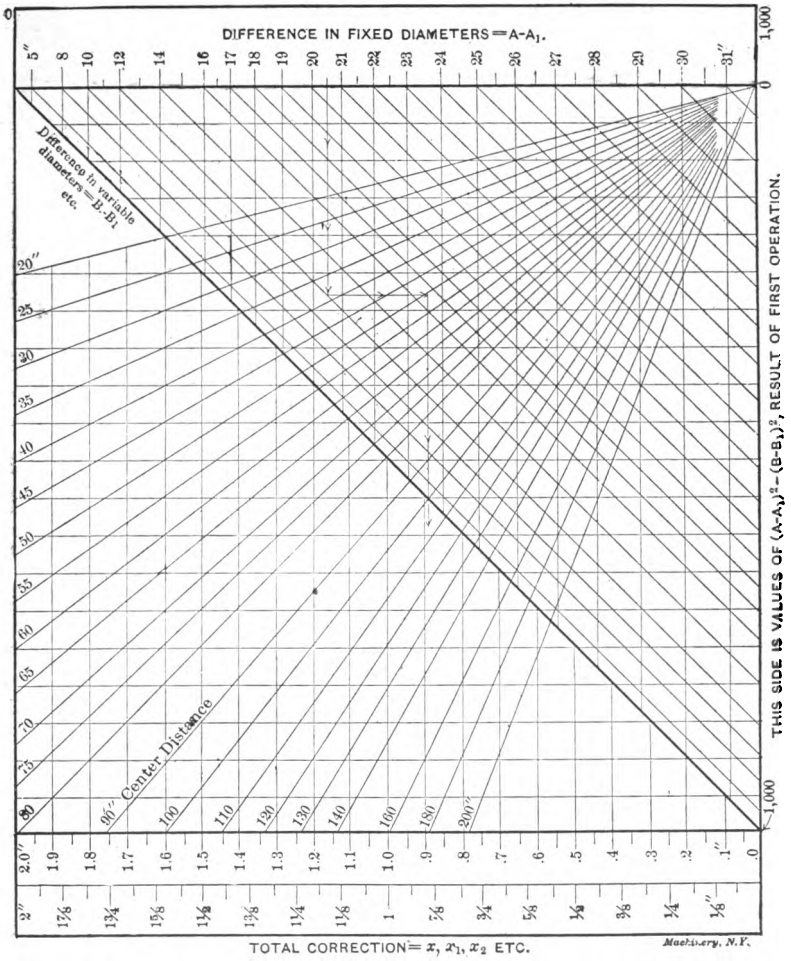
DIFFERENCE IN FIXED DIAMETERS = A-A1.
Difference in variable diameters = B-B1 etc.
Center Distance
THIS SIDE IS VALUES OF (A-A1)2 - (B-B1)2, RESULT OF FIRST OPERATION.
TOTAL CORRECTION = x, x1, x2 ETC.
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Fig. 10. Chart for Calculating Cone Pulleys.

which is the required answer. The range of this scale has been chosen from 0 to 2 inches, as this is probably the limit to which one could trust to the accuracy of the chart, since 2 inches increase in the sum of the diameters would make approximately 6 inches increase in belt length. Choosing the intersection of the graduated upper scale with

the zero point of the side scale as a point of origin, draw diverging lines to perform the last operation which is one of division. While these lines have a value of $6.283L$, they are designated by the values of L, the constant affecting the final value only.

Example in a Pair of Cones, Fig. 11.

$A = 26.25$ inches,
$B = 22$ inches,
$A_1 = 5.75$ inches,
$B_1 = 10$ inches,

as calculated for crossed belts according to the formula $A + A_1 = B + B_1$, the ratio of speed reduction being fixed by other conditions. $L = 50$ inches, $A—A_1 = 26.25 - 5.75 = 20.5$ inches, $B—B_1 = 22 - 10 = 12$ inches; correction $x = \dfrac{20.5^2 - 12^2}{6.283 \times 50} = 0.879$.

Solving this same problem by Chart Fig. 10, follow as indicated by the arrows; the result is read off at the bottom as 0.88 inches, which is near enough for all purposes. If the pair of cones are both to be cast

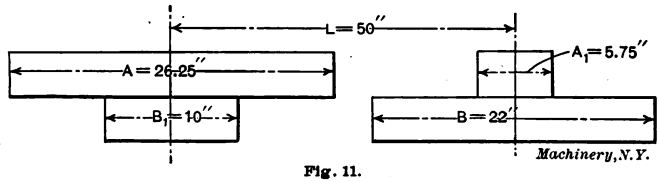


Fig. 11.

from the same pattern, the corrections need only be carried out up to and including the middle step, as the remaining steps are obviously duplicates of the already corrected ones.

Charts for Belt Transmissions.

The charts given in Figs. 12 and 13 will prove of value in calculations of transmissions. Fig. 12 is for the approximate determination of the width of belt to transmit a given horse-power at any stipulated speed. The formula upon which the chart is based is:

$$W = \frac{33{,}000 \times \text{H.P.}}{\frac{T}{2} \times V}$$

where W = width in inches,
T = working tension in belt,
V = velocity in feet per minute.

The three variable factors are H.P., $\frac{1}{T}$, and $\frac{1}{V}$. The above formula is based upon a coefficient of friction of 0.3 and an arc of contact of 135 degrees approximately, and takes no account of centrifugal action, which considerably reduces the effective belt tension at high velocities. We are only concerned with the constants in fixing the limits of the chart.

In plotting this chart, maximum and minimum values are taken as given in the accompanying table:

	H. P	T	V
Maximum	100	180	5,000
Minimum	10	60	500

Horse-power values are plotted to any suitable scale on the left of the diagram (in this figure carried to 120), values of $\frac{33{,}000 \times \text{H.P.}}{V}$ along the top line, and from the zero of the horse-power scale sloping velocity lines are drawn through the correct points upwards. On the right hand of the chart, downward, a scale of widths is plotted,

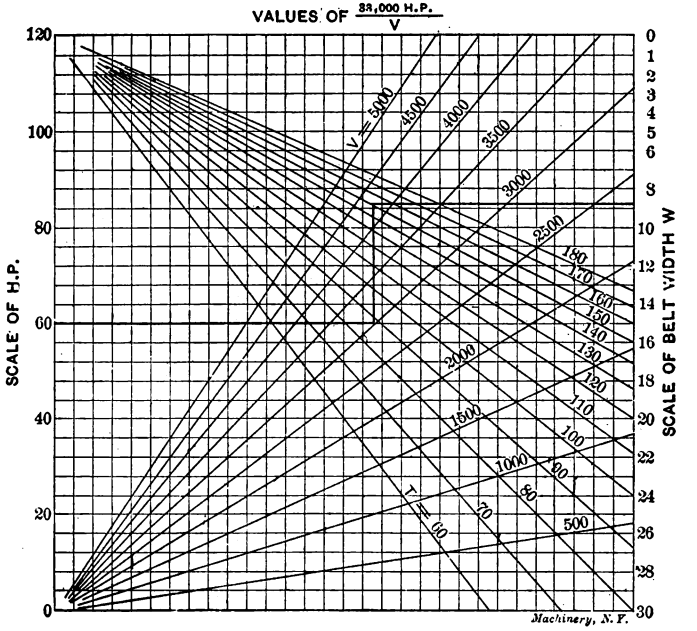


Fig. 12. Chart of Horse-power Transmitted by Belts.

reading to 30 inches. From the zero of the top scale sloping lines are drawn downward through the correct points for the several belt tensions.

As an example of the use of this chart, let us find the width of belt necessary to transmit 60 horse-power at a belt speed of 3,000 feet per minute, with an effective tension of belt of 150 pounds per inch of width. Enter the chart at 60 on the horse-power scale, and move along the heavy line in the figure to the intersection of the 3,000 velocity line, thence vertically to the intersection of the 150 tension line, and on the scale of widths find the required width of 9 inches. Obviously the process may be reversed, and given width, speed, and stress per inch width, we may find the horse-power transmitted.

As the width varies directly as the horse-power, it is obvious that the chart is applicable to any values within a reasonable limit, as by multiplying or dividing any horse-power on the chart by a multiple of ten, the corresponding width will be the width on the chart with the decimal point correspondingly shifted, and *vice versa.*

Fig. 13 is a chart expressing the relation between revolutions per

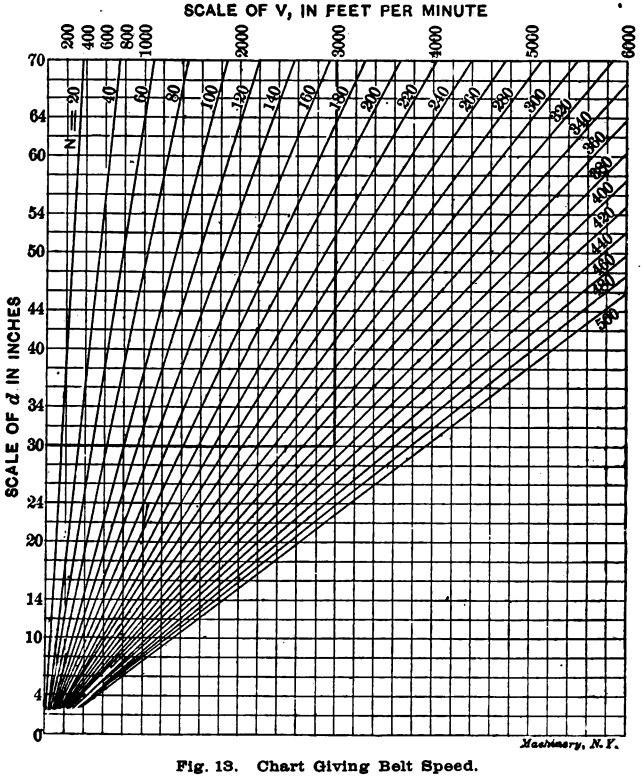


Fig. 13. Chart Giving Belt Speed.

minute, diameter of pulley in inches, and velocity of periphery in feet per minute, or the relation $V = \frac{\pi dN}{12}$, where N represents revolutions. This chart is drawn to read a maximum of 6,000 feet per minute. Following the heavy line, we find that a pulley 30 inches in diameter, running at 380 revolutions per minute, has a peripheral speed of 3,000 feet per minute. This chart may well be used in connection with Fig. 12.

CHAPTER II.

THE DESIGNING OF MACHINE FRAMES.

For men of limited experience, a machine frame of the common C type such as is used in punching and shearing machines, in some kinds of riveters, and, in a modified form, in slotters, drill presses, steam hammers, and many other tools, is always more or less difficult to design. The stresses are rather complex, and the means for securing the best distribution of metal for bearing these stresses is not easy. Much has been written on the subject, and the author of this chapter does not claim any originality in the ideas involved in the discussion. It is believed, however, that the method of presenting the solution of the problem has some points that may merit consideration. A common punching and shearing machine frame will be used for illustration, because the form and proportions of these frames are so

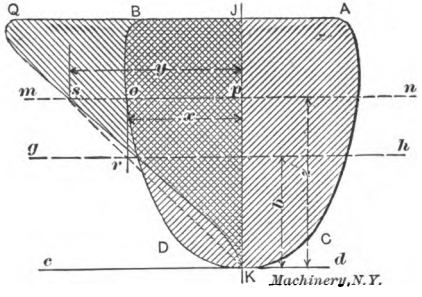


Fig. 14. Graphical Method for Finding the Center of Gravity.

well known. The method employed, however, is perfectly general and may be used for many other machine frames.

In the discussion which will follow it will be necessary to use the moment of inertia and center of gravity of an irregular section. The method of finding these quantities which is outlined below is not original with the author, but it is added here in the hope that its use may become more general.

Center of Gravity and Moment of Inertia of Irregular Figures.

The following graphical method for finding the center of gravity and moment of inertia of an irregular figure will be found to offer an advantage over the ordinary method by calculation, in the matter of time saved.

Let *ABDC*, Fig. 14, represent any irregular section. In the figure a section symmetrical about a vertical axis is shown, and the constructions carried out for half of the figure. For an unsymmetrical section both sides would have to be considered, as well as a construction for the gravity axis parallel to two base lines. A case of this sort will rarely arise in practice.

Lay off the line gh at any distance b from the edge of the section. The line gh is perpendicular to the center line JK. Then lay off another line mn, parallel to gh at any distance z from cd. Let the distance op be called x. From o drop a perpendicular on gh cutting at r. Join K and r and extend the line until it cuts mn at s, and let $sp = y$. After determining several points s in the same way connect them by a smooth curve, giving the figure KQJ.

Then by our construction we have,

$$\frac{x}{y} = \frac{b}{z}; \quad y = \frac{xz}{b}$$

The area of KQJ is

$$G = \int y dz = \frac{1}{b} \int xz dz$$

But $\int xz dz$ is the "static moment" of the original figure $KDBJ$, and therefore bG is the static moment of $KDBJ$ about cd. If A is the area of $KDBJ$

$$\frac{bG}{A} = \frac{\text{static moment of } KDBJ}{\text{area of } KDBJ}$$

$$= \text{distance from } cd \text{ to center of gravity} = e.$$

If the areas A and G and the distance b are measured, the distance to the center of gravity is readily found from this last equation. A planimeter offers the best means of measuring the areas, but if this is not possible, sufficient accuracy may be attained by dividing the areas to be measured into rectangles and triangles, computing their separate areas and taking the sum of the areas of all the subdivisions as the area of the original figure.

There is a similar construction for finding the moment of inertia, which should be carried out on the same figure as was used for finding the center of gravity. The two constructions are separated here in order to avoid confusion in explaining them.

In Fig. 15 the lines gh and mn are laid off as before so that pq is x, and pw is y. Then we is drawn from w perpendicular to gh; join K and e and extend Ke to f, calling $pf = y_1$. Several points f will give the figure KSJ, and we have, as before,

$$\frac{x}{y} = \frac{b}{z}; \quad y = \frac{xz}{b}$$

and also,

$$\frac{y_1}{y} = \frac{z}{b}; \quad y_1 = \frac{yz}{b} = \frac{xz^2}{b^2}$$

Then the area of KSJ is

$$J = \int y_1 dz = \frac{1}{b^2} \int xz^2 dz$$

but the moment of inertia

$$I = \int xz^2 dz$$

from which $$J = \frac{1}{b^2} I, \text{ and } I = b^2 J.$$

In this equation we know b and can measure J as before.

This construction gives the moment of inertia about the line cd, whereas it is generally required about the gravity axis. It might have been so found if the base line had been the gravity axis instead of the line cd. However, the mathematical computation for the transformation of axes is easily made. The following formula should be used:

$$I_o = I_{cd} - e^2 A,$$

where I_o = moment of inertia about gravity axis,
I_{cd} = moment of inertia about cd,
e = distance from cd to center of gravity,
A = area of $KDBJ$.

It is to be clearly understood that the values of e and I found above

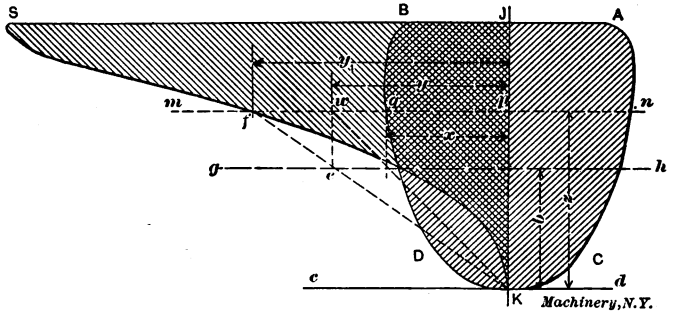


Fig. 15. Graphical Method for Finding the Moment of Inertia.

refer only to the figure actually drawn on the drawing board and not to the full-sized section. For the actual values we have

$$I_{\text{actual}} = X^4 I_{\text{drawing}}$$
$$e_{\text{actual}} = X e_{\text{drawing}}$$

where X is the scale of the drawing.

Fig. 21 shows the construction applied to a punch frame section.

The Stresses Involved in a Punch Frame.

Let Fig. 16 represent the side view of a punch frame. It will be sufficient to investigate the stresses at four different sections, testing at other points later if it appears necessary. The preliminary calculations will be made on sections indicated at AB, CD, EF, and GH or JK.

For determining the stresses on the section AB, consider the upper portion of the punch a "free body" as shown in Fig. 17. It is subjected to an external force, P, acting upwards, arising from the resistance of the material being punched. In the actual punch this force tends to separate the jaws. To balance P there is an internal stress acting over the section AB. The following discussion will show that this internal stress is a compound stress.

Conditions of equilibrium require that the algebraic sum of all

forces, external and internal, acting on a body must be zero; and also that the sum of all couples, or forces tending to produce rotation, must be zero. It is then evident that to balance P there must be downward forces equal to P. Such downward forces exist in what may be considered as a uniform tensile stress of t_1 pounds per square inch over the section AB. The resultant of all these small downward forces making up the uniform tensile stress can be considered as a single force, P_1, equal to P, acting at the center of gravity of the section AB. The algebraic sum of all the forces on the free body is now zero, but P and P_1 together form a couple of value $Pl_1 = P_1 l_1$, tending to rotate the portion of the punch under consideration in a clockwise direction, so that an equal and opposite internal couple is necessary to balance

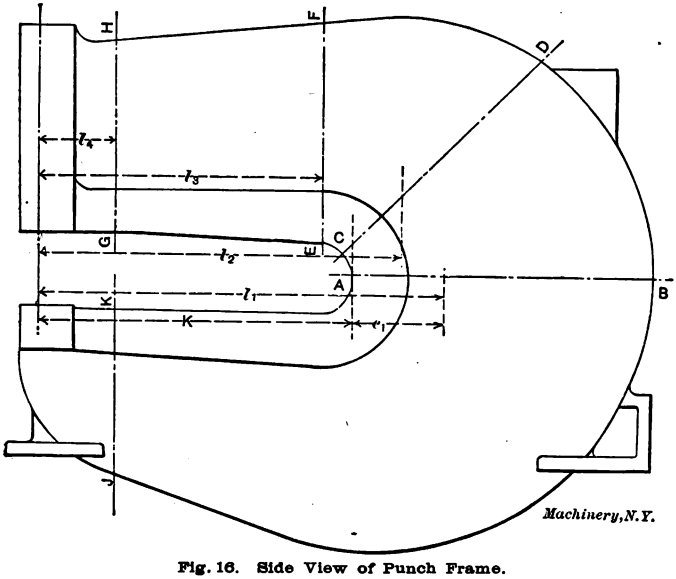


Fig. 16. Side View of Punch Frame.

it. This internal couple acts over the entire surface of the section AB, and may be considered as being made up of an infinite number of small couples, each consisting of a tensile stress on the throat side of the frame section and an equal compressive stress on the outer part of the section. (See Fig. 17.) Then the total internal stress may be found by combining these elementary stresses as shown.

The following formulas are used to calculate the values of these internal stresses. Their derivation is comparatively simple, and may be found in any treatise on the mechanics of materials.

$$P = \pi dts, \qquad (1)$$

where d = greatest diameter of hole to be punched on the machine considered,

t = thickness of plate punched,

s = shearing strength of material punched.

$$t_1 = \frac{P_1}{A_1} = \frac{P}{A_1}, \qquad (2)$$

where t_1 = uniformly distributed tensile stress over section AB, in pounds per square inch,

A_1 = area of section AB, in square inches.

$$f_1 = \frac{P_1 l_1}{N_1} = \frac{M_1 e_1}{I_1} \qquad (3)$$

where f_1 = tensile stress at point A, due to couple $P_1 l_1$, in pounds per square inch,

l_1 = arm of couple in inches $= K + e_1$ (Fig. 17),

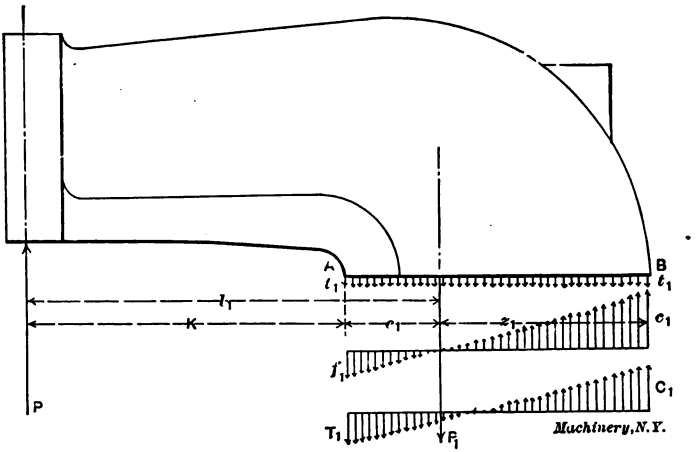


Fig. 17. Graphical Illustration of Stresses in Punch Frame.

e_1 = distance from center of gravity of section to edge next to punch throat, in inches,

N_1 = section modulus, for tension, of section $AB = \frac{I_1}{e_1}$,

M_1 = bending moment in inch-pounds,

I_1 = moment of inertia of AB about gravity axis.

$$c_1 = \frac{P_1 l_1}{Z_1} = \frac{M_1 z_1}{I_1} \qquad (4)$$

where c_1 = compressive stress, in pounds per square inch, at point B, due to couple $P_1 l_1$.

Z_1 = section modulus of section AB, for compression, $= \frac{I_1}{z_1}$

where z_1 is the distance from the center of gravity to the outer edge of the section.

The following equations may be written for the maximum stresses on the section AB.

Maximum unit tension

$$= T_1 = t_1 + f_1 = \frac{M_1 e_1}{I_1} + \frac{P_1}{A_1} = \frac{P_1 l_1 e_1}{I_1} + \frac{P_1}{A_1} \qquad (5)$$

Maximum unit compression

$$= C_1 = c_1 - t_1 = \frac{M_1 z_1}{I_1} - \frac{P_1}{A_1} = \frac{P_1 l_1 z_1}{I_1} - \frac{P_1}{A_1} \qquad (6)$$

In equations (5) and (6) everything is known from the conditions that would be given in the problem, as it is received by the designer, except the quantities $\frac{c_1}{I_1}$, $\frac{z_1}{I_1}$ and A_1. It will be seen that these are all factors depending on the size and shape of the section AB under consideration, and these are for the designer to determine.

Method of Designing Frame.

After clearly understanding the stresses involved in the design, the next step will be to proportion the frame to withstand these stresses. This may be accomplished in the following manner:

1. Assume the shape of the section at AB, Fig. 16. In making this assumption the designer will have to be guided by experience, either his own or that of others as shown in machines already built, catalogs, textbooks, drawings, photographs, or any other available means of gaining information that is available. The section shown in Fig. 21 will give a fairly good idea of the commonest form used for punches. In making the first drawing it is not necessary to make any assumption as to the *size* of the finished section, the *shape* is all that is necessary.

2. Lay out this section on a drawing board. Use any scale, and make the drawing rather large, say eight inches as the longest dimension. The *size* is of no particular importance, only be careful to get the figure drawn of the same proportions as desired in the final section.

3. Find the gravity axis and the moment of inertia of the figure drawn, by the graphical method explained above.

4. It is evident that the figure drawn will not be the correct *full sized* section of the punch, but it will be a drawing of that section to some scale. Let that unknown scale be such that one inch on the drawing represents X inches on the actual section. If, then, quantities relating to the figure on the drawing board are represented by letters with the subscript d, and quantities relating to the actual section by letters without additional subscripts, the following equations will hold true:

$$e_1 = X(e_1)_d \qquad (7)$$

$$z_1 = X(z_1)_d \qquad (8)$$

$$A_1 = X^2(A_1)_d \qquad (9)$$

$$I_1 = X^4(I_1)_d \qquad (10)$$

We can measure $(e_1)_d$, $(z_1)_d$, $(A_1)_d$, and $(I_1)_d$ directly from the draw-

ing, and insert the values for e_1, z_1, A_1, and I_1 in equations (5) and (6). We then have

$$T_1 = \frac{P_1 l_1 X(e_1)_d}{X^4(I_1)_d} + \frac{P_1}{X^2(A_1)_d} = \frac{P_1[K + X(e_1)_d]X(e_1)_d}{X^4(I_1)_d} + \frac{P_1}{X^2(A_1)_d} \quad (11)$$

$$C_1 = \frac{P_1 l_1 X(z_1)_d}{X^4(I_1)_d} - \frac{P_1}{X^2(A_1)_d} = \frac{P_1[K + X(e_1)_d]X(z_1)_d}{X^4(I_1)_d} - \frac{P_1}{X^2(A_1)_d} \quad (12)$$

Equation (11) can now be used to solve for X, if the allowable value for T_1 is known. A machine of the punch or shear type is subject to severe shock at the moment the tool strikes the metal. For this reason a very low allowable safe stress should be taken. It is suggested that the maximum allowable tensile or shearing stress be not greater than 2,000 to 2,200 pounds per square inch for cast iron. Equation (11) is a 4th power equation, and the quickest and easiest way to solve it is by trial. After finding the value of X from equation (11), it can be substituted in equation (12) to find the value of C_1. This should not exceed 8,000 to 10,000 pounds per square inch. It will usually be found much less than this, because mechanical difficulties are encountered in the manufacture of the punch if the outer part of the section is made thin enough to raise the compressive stress to the highest allowable value. These difficulties are mainly in the shape of extremely high cooling strains in the casting, so that the risk is too great to compensate for the comparatively small saving in iron. Take the value of X found and apply it to the original drawing to find the actual size of section. All linear dimensions of the drawing will be multiplied by X to get the actual size. This will fully determine section AB.

Having section AB, there are two methods of determining other sections. They may be found in a manner exactly similar to that used for section AB, by assuming a *shape* of section and solving for the *size*, or the size and shape may both be assumed and then tested for safety. The latter method will undoubtedly give the most satisfactory results, because it would probably require a great many trials by the first method for the reason that while each section would be strong enough to bear the stresses upon it, there would be considerable difficulty in combining these independently found sections into a pleasing and harmonious whole.

After finding section AB, use it as a basis, together with all the other known data about the punch, from which to assume an outline of the side view. In this part of the work, as in the first assumption of the shape of section AB, reference must be made to the results of past experience, and here again books, drawings, and illustrations will be found helpful. Fig. 16 is a side view of the frame of a punch made by one of the largest manufacturers of machine tools in the United States; if nothing better is at hand, it may be used as a guide. The thickness of the outer part of the frame is kept the same as was found for the section AB, and the frame is a trifle narrower from side to side at the tool end than at the back end.

Now, consider a section along some plane EF, parallel to the line of action of the tool. The "free body" is shown in Fig. 18. The forces acting on the free body are found by a method very similar to that used in discussing the stresses in the section AB. Let all quantities relating to this section be denoted by the subscript 3, and have the same meaning as the same letters had when used with subscript 1 in the discussion of section AB. The following equations may then be used to determine the stresses on the section.

P = the punching pressure.

P_3 = a force equal and opposite to P, causing shearing stress along EF.

P and P_3 with arm l_3 form a couple. This tends to produce bending

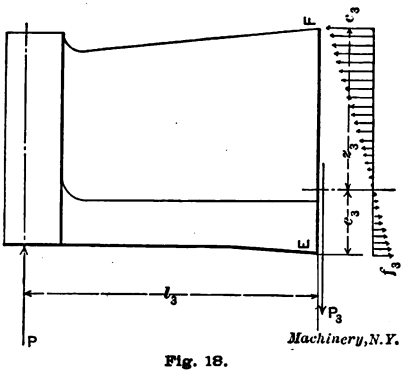


Fig. 18.

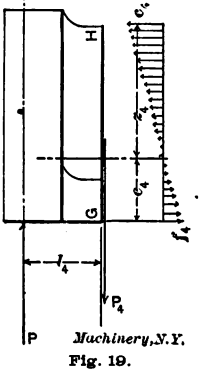


Fig. 19.

Graphical Illustrations of Stresses in Punch Frame.

in the frame and must be resisted by an internal moment of equal magnitude, M_3.

$$M_3 = P_3 l_3 = P l_3$$

As before, we have the tensile stress in section EF at E due to the couple as

$$f_3 = \frac{M_3 c_3}{I_3} \tag{14}$$

and the compressive stress in the section at F, due to the couple as

$$c_3 = \frac{M_3 z_3}{I_3} \tag{15}$$

The shearing stress is distributed uniformly over the entire section and its value is

$$s_3 = \frac{P_3}{A_3} = \frac{P}{A_3} \tag{16}$$

The maximum tension, compression, and shearing effects are found by combining f_3, c_3, and s_3 according to the rules found in various works on the mechanics of materials. "Kent's Mechanical Engineer's Pocket-Book" (7th edition, p. 283) gives the following rules, taken from Prof. Merriman's "Strength of Materials."

$$T_3 = \tfrac{1}{2}f_3 + \sqrt{s_3^2 + \tfrac{1}{4}f_3^2} \qquad (17)$$

$$C_3 = \tfrac{1}{2}c_3 + \sqrt{s_3^2 + \tfrac{1}{4}c_3^2} \qquad (18)$$

$$S_3 = \pm\sqrt{s_3^2 + \tfrac{1}{4}f_3^2} \quad \text{or} \quad \pm\sqrt{s_3^2 + \tfrac{1}{4}c_3^2} \qquad (19)$$

Since the value of s_3 is uniform over the entire section, while the stresses due to the couple are variable, it is evident that the maximum shearing effect will occur at that part of the section where the stresses due to the couple are a maximum, *i. e.*, probably at *F*.

The section must be designed strong enough to resist both tension and shear. In general, it is not possible to make it equally strong to resist both, when such a material as cast iron is used. At a section near the end, the shearing stress may be the one to fix proportions,

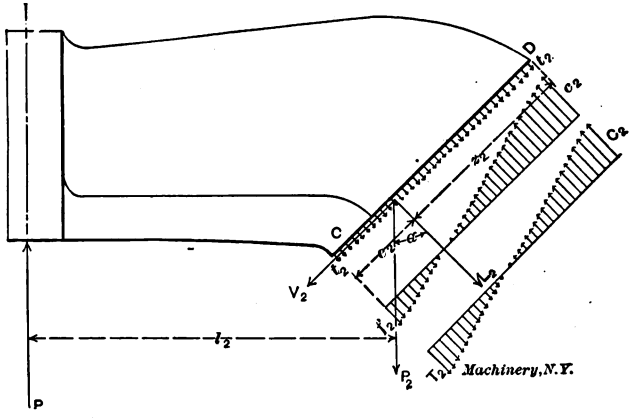


Fig. 20. Graphical Illustration of Stresses in Punch Frame

but when taken near the back of the throat, the tensile stresses are more liable to require consideration.

Equations (17), (18), and (19) should be solved for T_3, C_3, and S_3, and the results should show values within the maxima stated above. If the section is well designed; the values of T_3 and S_3 will not be very different from those mentioned, and as before, the section will probably be excessively strong in compression.

A section taken at *GH* or *JK* would be investigated in a manner exactly similar to that used for the section *EF*. (See Fig. 19.) It may be expected that the section at *GH* will appear much stronger than necessary, but this is because it is necessary to provide a space for the main shaft to pass through, as well as for the sake of appearance. The head will need to be large to accommodate the moving parts within it, and the part of the frame immediately behind the head must not appear too small to support the head.

It is next necessary to consider a section taken at an angle, for example along the line *CD*. The free body is shown in Fig. 20. The forces will be denoted by the same letters as heretofore, but with the subscript 2.

P = the punching pressure,

P_2 = a force equal and opposite to P; P_2 is resolved into two components, L_2 and V_2,

$L_2 = P_2 \cos \alpha$ (produces uniform tension),

$V_2 = P_2 \sin \alpha$ (produces shearing stress).

P and P_2 together form a couple which has a value $P_2 l_2 = P l_2$, which must be resisted by an internal moment, M_2.

$$M_2 = P_2 l_2 = P l_2. \tag{20}$$

As before,

$$f_2 = \frac{M_2 e_2}{I_2} \tag{21}$$

$$c_2 = \frac{M_2 x_2}{I_2} \tag{22}$$

$$t_2 = \frac{L_2}{A_2} \tag{23}$$

$$s_2 = \frac{V_2}{A_2} \tag{24}$$

The maximum stresses per square inch will then be

$$T_2 = \frac{f_2 + t_2}{2} + \sqrt{s_2^2 + \left(\frac{f_2 + t_2}{2}\right)^2} \tag{25}$$

$$C_2 = \frac{c_2 - t_2}{2} + \sqrt{s_2^2 + \left(\frac{c_2 - t_2}{2}\right)^2} \tag{26}$$

$$S_2 = \pm\sqrt{s_2^2 + \left(\frac{f_2 + t_2}{2}\right)^2} \text{ or } \pm\sqrt{s_2^2 + \left(\frac{c_2 - t_2}{2}\right)^2} \tag{27}$$

Take the greater of the two expressions in equation (27) for the subsequent work. This section will have dimensions between those of AB and EF.

Example of Calculations.

In order to illustrate the method of procedure outlined above, the following example is given. The machine in question is designed to punch a 1¼-inch hole in a 1-inch plate. The throat depth is 60 inches.

The punching force necessary may be found from equation (1).

$P = \pi dts = \pi \times 1.25 \times 1 \times 50{,}000 = 196{,}350$ pounds.

The punch frame is to be of the same general form as that shown in Fig. 16, except that the section at JK is to be of the same size as that at GH. By reference to various drawings and photographs the shape of the section at AB is assumed to be that shown in Fig. 21. This is drawn, in this case, on a drawing board, making the longest dimension 5.01 inches (for general practice it would be well to make the drawing larger than this, say 8 inches high), and the figures denoted by G_1 and J_1 constructed for finding the gravity axis and the moment of inertia. In this case only half of the construction is carried out, because the section is symmetrical about the vertical center line, and

it will be understood that the areas measured have to be doubled before they are used in the computations, *i. e.*, the values of the areas A_1, G_1, and J_1 used in the following calculations are not the areas actually measured from the drawing as shown in Fig. 21, but twice these values, to account for the construction for the right hand side of the section which is not carried out on the drawing board. With a value of b_1 as 1.98 inch, the areas of A_1, G_1, and J_1 when measured with a planimeter are as follows:

$A_1 = 3.84$ square inches,
$G_1 = 3.02$ square inches,
$J_1 = 4.72$ square inches.

Then the values of the moment of inertia and the distance from the

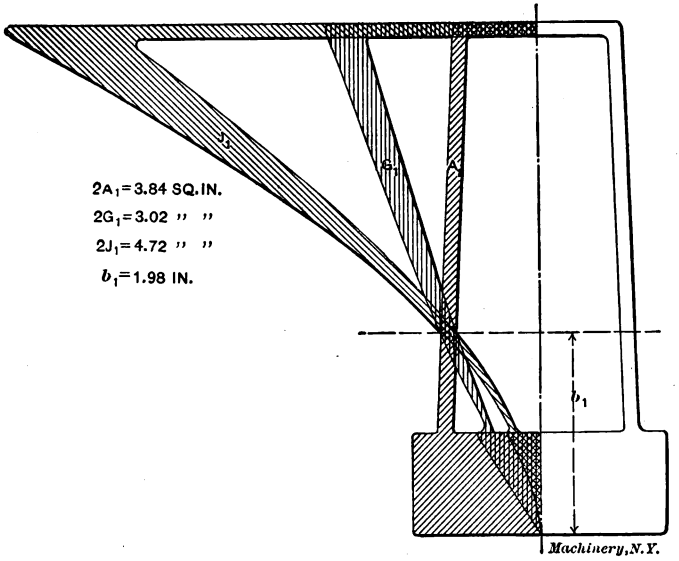


Fig. 21. Section of Punch Frame, showing Method of Determining the Center of Gravity and the Moment of Inertia.

bottom of the figure to the gravity axis may be found through the use of equations given in the first part of this chapter, as follows:

$$(e_1)_d = \frac{bG_1}{A_1} = \frac{1.98 \times 3.02}{3\ 84} = 1.56$$

$$(z_1)_d = 5.01 - (e_1)_d = 5.01 - 1.56 = 3.45$$

$$(I_1)'_d = b^2 J_1 = (1.98)^2 \times 4.72 = 18.5$$

$$(I_1)_d = (I_1)'_d - (e_1)^2_d A_1 = 18.5 - (1.56)^2 \times 3\ 84 = 9.2$$

These values apply only to the figure which was drawn on the drawing board and not to the actual section. It is next necessary to find the size of the actual section. This is accomplished by assuming T_1 as 2,000 pounds per square inch and solving for X in equation (11).

$$T_1 = \frac{P_1[K + X(e_1)_d]X(e_1)_d}{X^4(I_1)_d} + \frac{P_1}{X^2(A_1)_d} \qquad (11)$$

$$2000 = \frac{196{,}350\,[60 + 1.56\,X]\,1.56\,X}{X^4 \times 9.2} + \frac{196{,}350}{X^2 \times 3.84}$$

from which the value $X = 11.45$ is found by trial.

Therefore, to get the actual size of the section at *AB*, each dimension of the drawing is multiplied by 11.45. This gives the depth of the section as 57.36 inches, and also

$$e_1 = 1.56 \times 11.45 = 17.86 \text{ inches},$$
$$z_1 = 3.45 \times 11.45 = 39.5 \text{ inches},$$

and all other dimensions in the same proportions.

The value of C_1 is next calculated by equation (12) and found to be 3,420 pounds per square inch. It would seem that this is very low

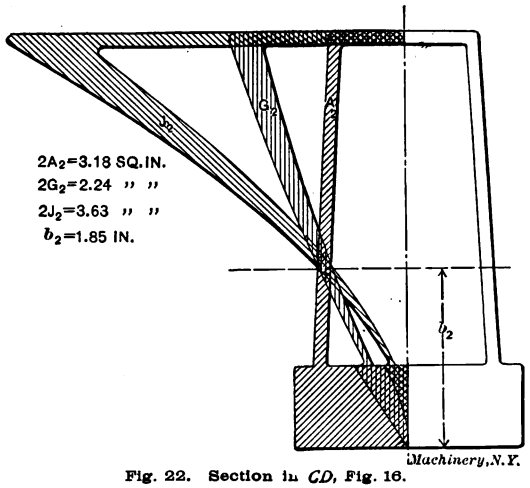


Fig. 22. Section in *CD*, Fig. 16.

for a compressive stress, but the thickness of the outer part of the frame is only about 1⅝ inch, and it does not seem best to make it thinner.

After having determined the size and shape of the section *AB*, the outline of the punch is sketched in, the general shape being about as shown in Fig. 16, and care is taken to secure a form of pleasing appearance and conforming to general practice. Then drawings are made of sections at *CD*, *EF*, *GH*, and *JK*, as shown in Figs. 22, 23, and 24. using a scale $X = 12$. The constructions for finding e and z are completed as shown, and the results of the calculations for the sections are tabulated below:

Section	A	G	J	b	e	z	I'	I
CD	3.18	2.24	3.63	1.85	1.304	2.976	12.44	7.03
EF	2.74	1.83	2.58	1.56	1.04	2.51	6.47	3.51
GH }	2.26	1.74	2.47	1.27	0.978	2.02	3.98	1.82
JK }								

The next step in the design is to investigate the stresses in the sections at *CD*, *EF*, *GH*, and *JK*. The calculations are shown below. All calculations are conveniently made with a slide rule.

Section *CD*.

This section is taken at an angle $\alpha = 45°$. Then, from equations developed in the discussion of the section *CD* in the earlier part of this chapter, we have:

$$P_2 = P = 196{,}350 \text{ pounds.}$$
$$L_2 = P_2 \cos \alpha = 196{,}350 \times 0.707 = 138{,}700 \text{ pounds.}$$
$$V_2 = P_2 \sin \alpha = 196{,}350 \times 0.707 = 138{,}700 \text{ pounds.}$$
$$l_2 = \text{(by measurement from drawing) } 69.25 \text{ inches.}$$
$$M_2 = P_2 l_2 = 196{,}350 \times 69.25 = 13{,}600{,}000 \text{ inch-pounds.}$$
$$f_2 = \frac{M_2 e_2}{I_2} = \frac{13{,}600{,}000 \times 12 \times 1.304}{12^4 \times 7.03} = 1460 \text{ pounds.}$$

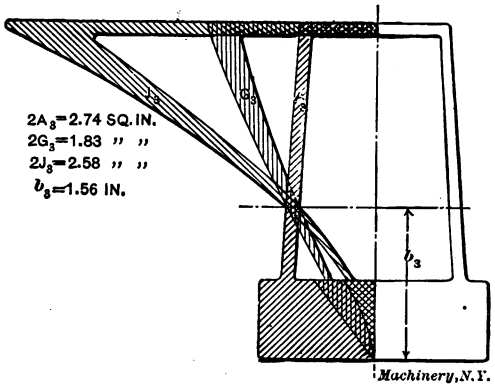


Fig. 23. Section in *EF*, Fig. 16.

$$c_2 = \frac{M_2 z_2}{I_2} = \frac{13{,}600{,}000 \times 12 \times 2.976}{12^4 \times 7.03} = 3326 \text{ pounds.}$$
$$t_2 = \frac{L_2}{A_2} = \frac{138{,}700}{12^2 \times 3.18} = 303 \text{ pounds.}$$
$$s_2 = \frac{V_2}{A_2} = \frac{138{,}700}{12^2 \times 3.18} = 303 \text{ pounds.}$$
$$T_2 = \frac{1460 + 303}{2} + \sqrt{303^2 + \left(\frac{1460 + 303}{2}\right)^2} = 1811 \text{ pounds.}$$
$$C_2 = \frac{3326 - 303}{2} + \sqrt{303^2 + \left(\frac{3326 - 303}{2}\right)^2} = 3056 \text{ pounds.}$$
$$S_2 = \sqrt{303^2 + \left(\frac{3326 - 303}{2}\right)^2} = 1544 \text{ pounds.}$$

It will be noted that the tensile stress is somewhat smaller than the

maximum value of 2,000 pounds per square inch which was to be allowed. However, the strength of the casting is likely to be least at this point for two reasons; first, because the section is taken at the center of a rather sharp bend for so massive a casting, and second, because of the necessity of introducing a bearing on the upper side near this section, thus requiring the frame to support the reaction from the bearing. For these reasons a lower value for the tension is not out of place.

Section *EF*.

If the tabulated values of the properties of the section *EF* are substituted in the equations developed for that section, the following values of the stresses result:

$P_3 = P = 196{,}350$ pounds.

$l_3 = 54$ inches (measured from drawing).

$M_3 = P_3 l_3 = 196{,}350 \times 54 = 10{,}600{,}000$ inch-pounds.

$$f_3 = \frac{M_3 e_3}{I_3} = \frac{10{,}600{,}000 \times 12 \times 1.04}{12^4 \times 3.51} = 1815 \text{ pounds.}$$

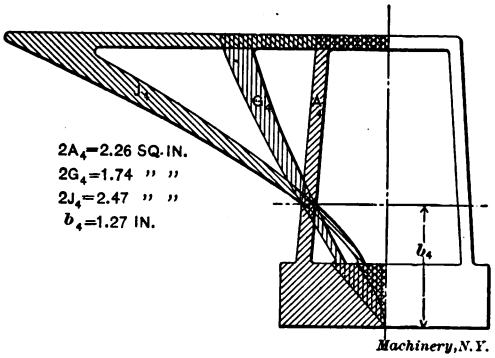


Fig. 24. Section in *GH* and *JK*, Fig. 16.

$$c_3 = \frac{M_3 z_3}{I_3} = \frac{10{,}600{,}000 \times 12 \times 2.51}{12^4 \times 3.51} = 4380 \text{ pounds.}$$

$$s_3 = \frac{P_3}{A_3} = \frac{196{,}350}{12^2 \times 2.74} = 498 \text{ pounds.}$$

$$T_3 = \frac{1815}{2} + \sqrt{498^2 + \left(\frac{1815}{2}\right)^2} = 1944 \text{ pounds.}$$

$$C_3 = \frac{4380}{2} + \sqrt{498^2 + \left(\frac{4380}{2}\right)^2} = 4438 \text{ pounds.}$$

$$S_3 = \sqrt{498^2 + \left(\frac{4380}{2}\right)^2} = 2248 \text{ pounds.}$$

These results are all fairly near the requirements, and the section may be said to be satisfactory. The value of S_3 is a trifle higher than

was originally specified; this could be reduced by increasing the area of the section slightly. The excess is, however, hardly enough to warrant this.

Sections *GH* and *JK*.

Sections *GH* and *JK* are practically alike in the punch under discussion, although they are not so drawn in Fig. 16. The section as shown in Fig. 16 at *GH* is nearer the size to be investigated than that at *JK*. The stresses will result as follows:

$$P_4 = P = 196{,}350 \text{ pounds}$$

$$l_4 = 14.5 \text{ inches (measured from drawing).}$$

$$M_4 = P_4 l_4 = 196{,}350 \times 14.5 = 2{,}850{,}000 \text{ inch-pounds.}$$

$$f_4 = \frac{M_4 e_4}{I_4} = \frac{2{,}850{,}000 \times 12 \times 0.978}{12^4 \times 1.82} = 885 \text{ pounds.}$$

$$c_4 = \frac{M_4 z_4}{I_4} = \frac{2{,}850{,}000 \times 12 \times 2.02}{12^4 \times 1.82} = 1830 \text{ pounds.}$$

$$s_4 = \frac{P_4}{A_4} = \frac{196{,}350}{12^2 \times 2.26} = 603 \text{ pounds.}$$

$$T_4 = \frac{885}{2} + \sqrt{603^2 + \left(\frac{885}{2}\right)^2} = 1189 \text{ pounds.}$$

$$C_4 = \frac{1830}{2} + \sqrt{603^2 + \left(\frac{1830}{2}\right)^2} = 2011 \text{ pounds.}$$

$$S_4 = \sqrt{603^2 + \left(\frac{1830}{2}\right)^2} = 1096 \text{ pounds.}$$

It is to be noted that in the case of these sections all the stresses are decidedly small. This is caused by the large section required at *GH* to provide room for the passage of the main shaft, and also because a smaller section at this point would present a weak appearance as a support for the massive head. In the case of the section at *JK*, however, these reasons do not hold, and the frame in question would have been better proportioned had the section at *JK* been smaller, thus making the stresses more nearly equal over the entire frame. The punch frame illustrated in Fig. 16 is better proportioned in this respect.

The system of design here presented may be applied to frames of many other forms of machines besides punching and shearing machines. It offers the following advantages:

1. It is mathematically correct in principle.
2. The sources of error are comparatively small and are of a kind easily detected.
3. The graphical constructions involved are not laborious, and are sufficiently exact for any ordinary work. The degree of accuracy can be increased at will by employing larger scales.
4. Considerable latitude is allowed for the judgment of the designer.

Physical Characteristics of Cast Iron.

In connection with the previous discussion regarding punch frames it may be appropriate to call attention to some characteristics of cast iron, as stated by Mr. James Christie in a paper read before the Engineers' Club of Philadelphia. Cast iron is probably the most complex, variable and uncertain form in which iron is used. Not only is the amount of extraneous metals and metalloids variable, but the condition in which the associated carbon exists, and the character of this association, are determined largely by the influence of silicon and possibly other metalloids. Again, the physical properties of the metal are influenced by casting temperature, rate of cooling, etc., so that altogether we can only assume the probable strength and stiffness of a

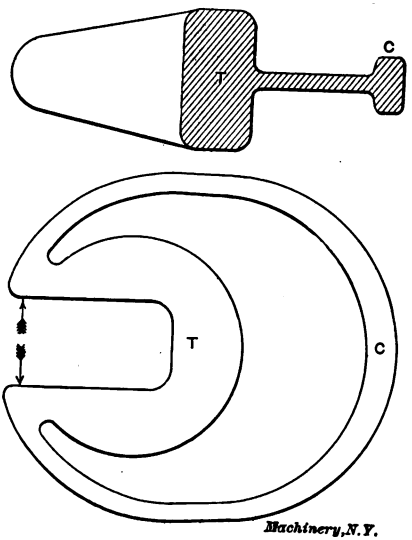


Fig. 25. Illustration of Common Design Intended to Equalize Stresses in Cast Iron.

casting in the most general way, and forecast results, which will suit an average, from which individual castings may vary widely in extremes.

For heavy machinery, where cast iron is used in heavy masses through which working stresses are imperfectly distributed, the metal probably is much softer and weaker in the middle of the mass, where it has cooled slowly, than at the outer surface, where the metal has more rapidly cooled. Furthermore, castings are usually under considerable internal strain, due to unequal contraction, and although this internal strain gradually disappears, it may have some disturbing influence after the casting has been put in service. It is good practice to assume an ultimate tensile strength of 16,000 pounds per square inch for ordinary iron castings, and to limit working stresses from

2,000 to 4,000 pounds per square inch, according to the conditions and character of the service.

Cast iron offers a high resistance to compressive stress, and although this resistance varies within wide limitations, it may be assumed as a working basis to be about six times that of the tensile strength, or, say, 95,000 pounds per square inch of section.

In the middle of the past century, as cast iron became extensively applied to structural purposes, its physical properties were studied with great care, and the experiments of Hodgkinson and Fairbairn in England, and of their contemporaries, yielded a valuable fund of information on the subject. Seeking a section of beam which should exhibit the highest ultimate strength in proportion to area of cross-section, or of the weight of metal employed, Hodgkinson advocated a section in which the tension flange exceeded the compression flange about six to one in sectional area, the web usually tapering in thickness from the tension flange, diminishing toward the other flange. This form of beam was largely adopted, and took precedence as long as cast iron was used for beams in structures. We find that the same method of reasoning influenced the machine designer in disposing cast iron to seeming advantage in the construction of machines, massing the metal to resist tension, and permitting high unit stress on metal in compression; and especially is this observed in machines of the open jaw or gap type, such as presses, punching and shearing machines, etc. It is believed by many good machine designers that usually the unit stresses should be little, if any, higher in compression than in tension, for the following reasons. In machinery, rigidity or stiffness is usually the chief consideration; many machines do not fulfil the intended purpose properly, not by failure through fracture, but by want of sufficient stiffness. Deflection has to be limited, and when that is done, breaking from excessive tension is sufficiently guarded against. As cast iron yields to compression as much as with the same unit stresses it yields to tension, it follows that the compressive stress should not exceed the tensile strength per unit of section if it is desired to dispose a given mass of metal with least deflection. It is believed that rupture sometimes occurs in a machine apparently through tension, where the origin of the weakness could be traced to a want of material to sufficiently resist compression, the improperly supported tension side severing by cross-bending or transverse stress.

Taking for illustration an open gap machines with frame as shown in Fig. 25, with tension at T and compression at C, if the section is so shaped that the compressive unit stress is six times that of the tensile unit stress, then, elastic moduli being equal, the frame will yield at C six times as much by compression as it does by tension at T. This permits an oscillation of the mass at T around its center. If this oscillation becomes dangerous, by extent or frequency, the frame will break by cross-bending at the mass T, giving the impression that more material is needed to resist tension, whereas the fact may be that more material should be placed at C to prevent excessive yield by compression.

When rapidly alternating stresses occur, it is acknowledged that provision must be made for something more than the greatest stress in one direction alone. There are still differences of opinion and practice on this subject among bridge designers. Some maintain that when the alternations are of slow recurrence, so as to permit actual rest between reversals, no special increase of section is required. Others specify that the sum of the sections required for the stress in opposite direction should be used to suit the conditions. There can be little doubt that the latter estimate is not too great for machinery when the oscillation of the forces occurs with great rapidity, and especially when the metal under consideration is cast iron, with a modulus of elasticity about one-half that of steel or wrought iron. It is a safe general rule for ordinary cast iron in machine construction to limit tensile stress to 4,000 pounds per square inch of section, under the most favorable circumstances; to 3,000 pounds when loads are suddenly applied; and to 2,000 pounds when the force alternates in direction; these unit loads are to be further limited to suit the ratio of length to section, as required for columns or any members in alternate extension or compression, or for beams or members subjected to alternating transverse stresses.

CHAPTER III.

BENDING STRESSES IN WIRE ROPE.

The study of different engineering problems brings out the proposition of two kinds of stresses, one seen and the other unseen. Those stresses which are caused by direct thrust or pull are easily seen and provided for, but the unknown forces, or the ones which fail to make an impression on our senses, are more difficult to grasp. Engineers have provided for such forces on some kinds of work by using a larger factor of safety. This method at best, however, only approximates, and should never be used except as a last resort, after all possible means for the determination of the stresses have been exhausted.

TABLE III. TOTAL AREA IN SQUARE INCHES OF WIRES OF DIFFERENT ROPES.

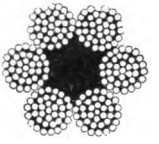
Diameter of Rope.	6 x 7 Construction.	6 x 19 Construction.	6 x 37 Construction.	8 x 19 Construction
2¾	.....	2.6892	2.6704	.....
2½	.....	2.2224	2.2068	.....
2¼	.....	1.8000	1.7876	.....
2	.....	1.4216	1.4124	.....
1¾	.....	1.0888	1.0812	.....
1⅝	..	0.9390	0.9324	.....
1½	0.8334	0.7997	0.7929	0.6714
1⅜	0.7007	0.6723	0.6676	0.5643
1¼	0.5791	0.5556	0.5517	0.4664
1⅛	0.4691	0.4500	0.4469	0.3778
1	0.3706	0.3554	0.3531	0.2985
⅞	0.2840	0.2722	0.2703	0.2285
¾	0.2134	0.1999	0.1982	0.1678
⅝	0.1448	0.1389	0.1379	0.1166
9/16	0.1173	0.1125	0.1117	0.0945
½	0.0926	0.0889	0.0883	0.0746
7/16	0.0710	0.0681	0.0676	0.0571
⅜	0.0534	0.0500	0.0495	0.0419
5/16	0.0362	0.0347	0.0345	0.0291
¼	0.0231	0.0222	0.0221	0.0186

Wire rope, on account of its numerous applications to various engineering problems, such as derricks, traveling cranes, elevators, ore- and coal-handling machinery, etc., is becoming a vital adjunct to the solution of the problem of the economical handling of many different materials. The draftsman or designer of any apparatus requiring the use of wire rope has to decide upon the size of sheaves and drums which he will employ, and in a great many cases has only a vague idea of what size these should be made. To be sure, the catalogues of wire rope manufacturers give in a general way the smallest diameter

of sheave or drum that should be used for any given size of rope, but it was never intended that sizes of sheave given as a minimum should be the only size to employ. Nevertheless, judged by this standard, a good many pieces of apparatus, of excellent design in other respects, show an almost total disregard for such recommendations, and the sheaves and drums used are of extremely small diameter. This is caused by a desire to make a compact apparatus, and to bring the first cost as low as possible, both of which are desirable features from a

Machinery, N.Y.

TABLE IV. STRENGTH OF 6x19 AND 6x37 ROPES.



Machinery, N.Y.

Diameter in Inches.	Approximate Circumference in Inches.	Weight per Foot in Pounds.	Approximate Breaking Stress in Tons of 2000 Pounds. Iron.	Approximate Breaking Stress in Tons of 2000 Pounds. Crucible Steel.	Approximate Breaking Stress in Tons of 2000 Pounds. Plow Steel.
2 1/2	8 1/2	11.95	114	228	305
2 1/4	7 1/2	9.85	95	190	254
2 1/4	7 1/2	8.00	78	156	208
2	6 1/4	6.30	62	124	165
1 3/4	5 1/2	4.85	48	96	128
1 5/8	5	4.15	42	84	111
1 1/2	4 1/2	3.55	36	72	96
1 3/8	4 1/4	3.00	31	62	82
1 1/4	4	2.45	25	50	67
1 1/8	3 1/2	2.00	21	42	56
1	3	1.58	17	34	44
7/8	2 3/4	1.20	13	26	34
3/4	2 1/4	0.89	9.7	19.4	25
5/8	2	0.62	6.8	13.6	18
9/16	1 3/4	0.50	5.5	11.0	14.5
1/2	1 1/2	0.39	4.4	8.8	11.4
7/16	1 1/4	0.30	3 4	6.8	8.85
3/8	1 1/8	0.22	2.5	5.0	6.55
5/16	1	0.15	1.7	3.4	4.50
1/4	3/4	0.10	1.2	2.4	3.00

business standpoint. The purchaser of such an apparatus, however, is also interested in the cost of maintenance as well as in the first cost. He finds after a short period that his ropes have all gone to pieces, and procures another rope only to get the same result. In the design of his machine too small sheaves were used, and he finds it necessary to replace, at considerable expense, the small sheaves by larger ones, or to continue to have a high cost of maintenance. An example of this will perhaps make this point clear. A 1¼-inch crucible steel rope 6 × 19 runs over a 3-foot sheave and drum on a machine designed to lift 10 tons. According to the tables furnished by rope

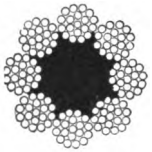
manufacturers, this rope has a strength of 50 tons, which would give a factor of safety of 5. We will assume that the factor 5 is as small as it is advisable to use for this particular service. What we actually have, however, is this:

Stress on rope due to load to be lifted..... = 10.00 tons
Stress on rope due to bending over 3-foot sheave..... = 7.25 tons

Total stress ..... = 17.25 tons

This gives a factor of safety of less than 3, which is altogether too small, showing how necessary it is to take into account the bending

TABLE V. STRENGTH OF 8x19 ROPES.



Machinery, N.Y.

Diameter in Inches.	Approximate Circumference in Inches.	Weight per Foot in Pounds.	Approximate Breaking Stress in Tons of 2000 Pounds. Crucible Steel.	Approximate Breaking Stress in Tons of 2000 Pounds. Plow Steel.
1 1/2	4 3/4	3.48	65	86
1 3/8	4 1/4	2.51	56	74
1 1/4	4	2.18	45	60
1 1/8	3 1/2	1.82	38	50
1	3	1.32	27	35
7/8	2 3/4	1.05	21	27
3/4	2 1/4	0.89	16.5	21
5/8	2	0.53	11.6	15
9/16	1 3/4	0.43	9.4	12.3
1/2	1 1/2	0.31	6.6	8.55
7/16	1 1/4	0.27	6.1	7.95
3/8	1 1/8	0.18	3.3	4.25
5/16	1	0.12	2.2	2.92
1/4	3/4	0.066	1.6	1.95

stress due to winding around a pulley or drum. It would be necessary to use in this case either a larger rope and larger sheaves and drums, or a stronger rope and sheaves and drums enough larger to reduce the bending stress to a smaller amount. A 1¼-inch plow steel rope, 6 × 19, has a strength of 67 tons, and with a factor of safety of 5 gives a working stress of 13.4 tons. The problem could then be solved as follows:

Stress on rope due to load to be lifted..... = 10.00 tons
Stress on rope due to bending over 6-foot 6-inch sheave... = 3.36 tons

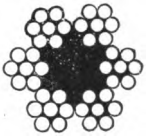
Total stress ..... = 13.36 tons

Note that the sheave had to be increased to more than double its original size to reduce the stress sufficiently. The bending stresses

given in the two cases outlined above are the actual stresses that any rope of the construction noted would have. The comparison above shows how vital it is that the bending stresses be carefully considered.

For the purpose of determining what the stress is, curves have been plotted for various kinds of commercial ropes of standard makes and various constructions. These curves, given in Figs. 26 to 33, show graphically the effect of the different sizes of sheaves on different ropes, so that it is possible for anyone to look at the curves and tell at a glance exactly what bending stress is put on the rope. For example, take the curves for 6 x 19 ropes, Figs. 28 and 29. There are

TABLE VI. STRENGTH OF 6x7 ROPES.



Machinery, N.Y.

Diameter in Inches.	Approximate Circumference in Inches.	Weight per Foot in Pounds.	Approximate Breaking Stress in Tons of 2000 Pounds. Iron.	Approximate Breaking Stress in Tons of 2000 Pounds. Crucible Steel.	Approximate Breaking Stress in Tons of 2000 Pounds. Plow Steel.
1 1/2	4 3/4	3.55	34	68	91
1 3/8	4 1/4	3.00	29	58	78
1 1/4	4	2.45	24	48	64
1 1/8	3 1/2	2.00	20	40	53
1	3	1.58	16	32	42
7/8	2 3/4	1.20	12	24	32
3/4	2 1/4	0.89	9.3	18.6	24
11/16	2 1/8	0.75	7.9	15.8	21
5/8	2	0.62	6.6	13.2	17
9/16	1 3/4	0.50	5.3	10.6	14
1/2	1 1/2	0.39	4.2	8.4	11
7/16	1 1/4	0.30	3.3	6.6	8.55
3/8	1 1/8	0 22	2.4	4.8	6.35
5/16	1	0 15	1.7	3.4	4.35
9/32	7/8	0.125	1.4	2.8	3.65

two sets of curves for this kind of rope, one plotted to show the relation between the diameter of rope and the bending stress (Fig. 29), and the other between the diameter of rope and the size of the sheave (Fig. 28). With any two factors known, the third can easily be obtained. Suppose that we desire to use a 1-inch diameter, 6 x 19 rope, of a strength of 34 tons to hoist a load of 4 tons. What is the minimum size sheave that can be used with a factor of safety of 5?

34 tons divided by 5 ..... = 6.8 tons total permissible stress
Direct load ..... = 4.0 tons

Difference ..... = 2.8 tons permissible bending stress

Using the diagram Fig. 29 we find that the line representing 2.8

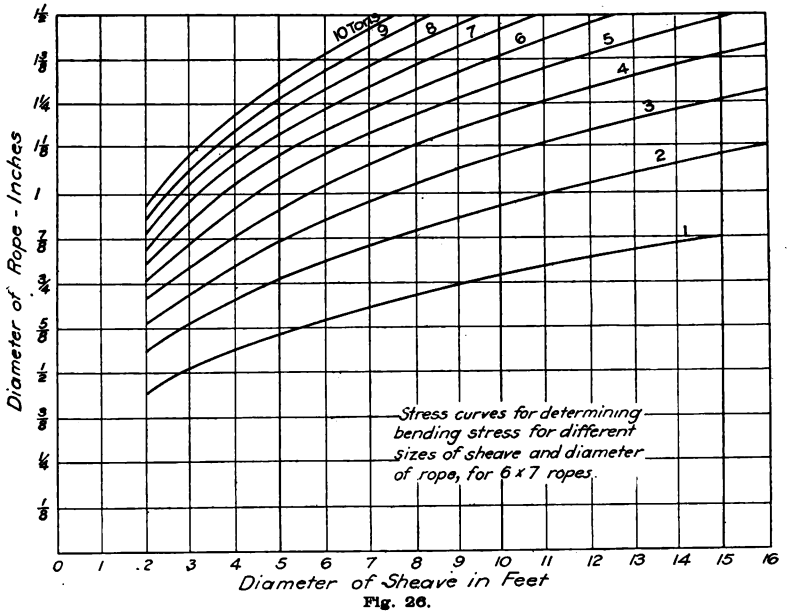


Fig. 26.

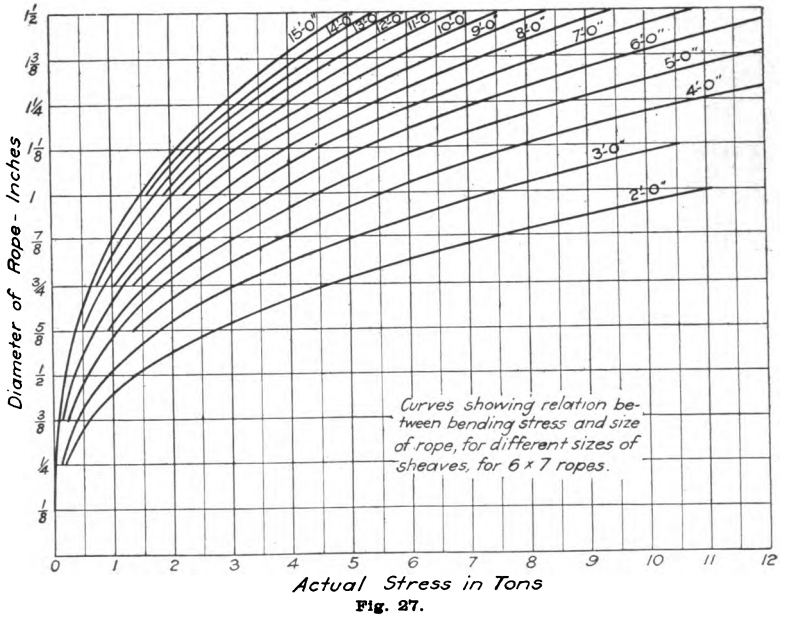


Fig. 27.

tons stress intersects the line for 1-inch rope on the curve marked 4 feet. Consequently we will require a sheave 4 feet in diameter.

We can reverse the question. A 6 x 19 rope, 1 inch in diameter, is

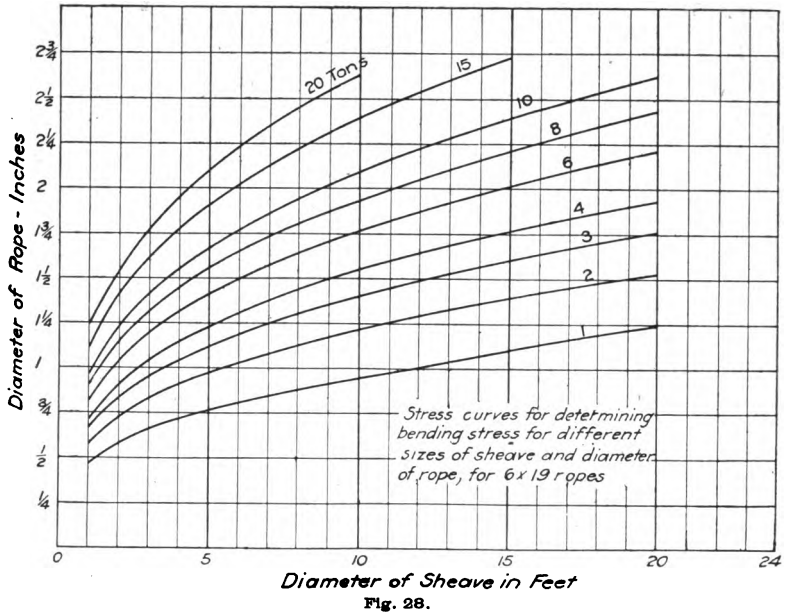


Fig. 28.

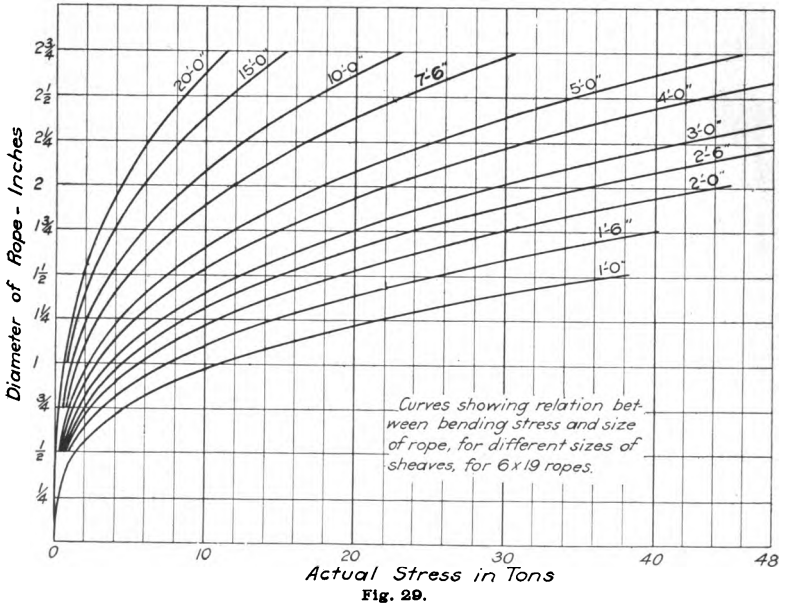


Fig. 29.

to be run over a 3-foot sheave; what load will it lift, assuming a factor of safety of 5 and a strength of rope given as 34 tons? Referring to the curves in Fig. 28, we find by following the line for 1-inch

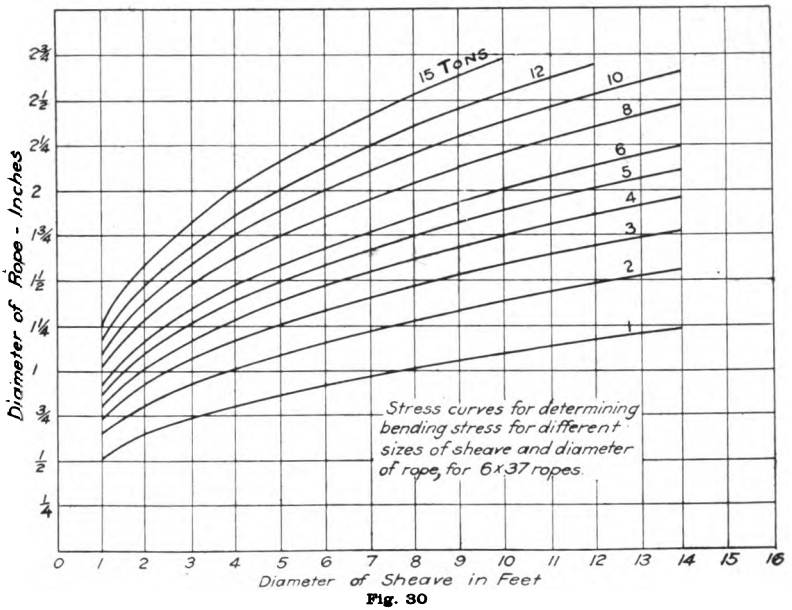


Fig. 30

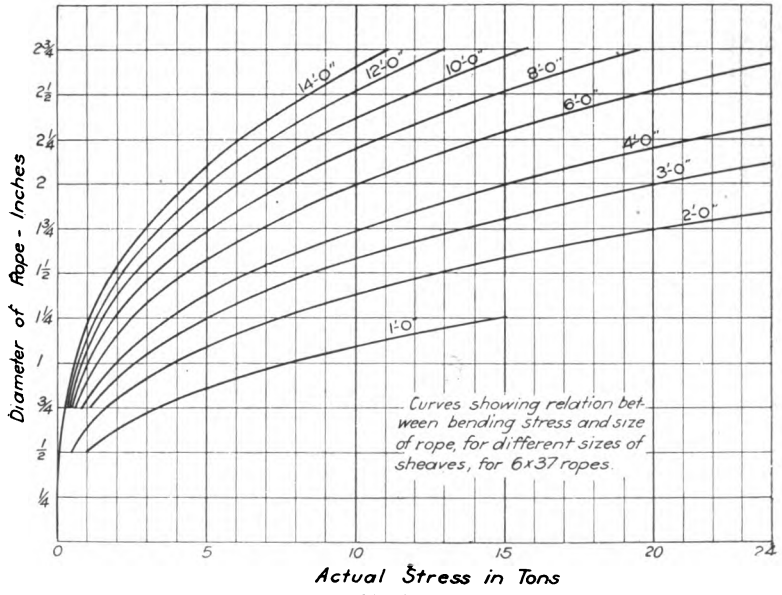


Fig. 31.

rope until it intersects the line for a 3-foot sheave that the bending stress lies between the curves for 3 tons and 4 tons. By proportioning, we find that it is about 3.7 tons.

We then have

34 tons divided by 5 ..... = 6.8 tons total permissible stress
Deducting ..... 3.7 tons bending stress

Difference ..... = 3.1 tons allowable working load.

Calculations to Obtain Curves.

The method of calculating these curves is as follows:

Let S = stress per square inch,

E = Young's modulus of elasticity of steel = 29,000,000,

E_r = modulus of elasticity of the rope as a whole,

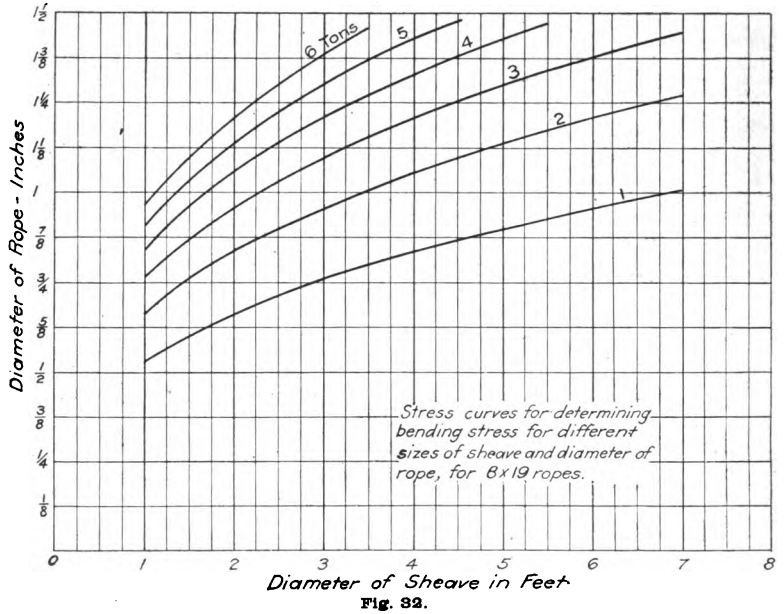


Fig. 32.

d = diameter of wire of rope in inches,

d_r = diameter of rope in inches,

D = diameter of sheave or drum in inches.

E_r is a function of E, that is, it is dependent upon its value. E_r is less than E on account of the structure of a rope, and varies with the flexibility of same. A low modulus is indicative of a flexible rope and *vice versa*. In considering this problem, we will take into account the modulus of elasticity E_r of the rope as a whole. This conception is different from the ordinary conception of the modulus of elasticity, but is perfectly feasible, as the modulus of elasticity is, strictly speaking, the ratio between the force applied to any material per square inch and the amount of elongation expressed as a fraction of the original length.

The stress produced in a round bar of the diameter d_1, when bent around a sheave of diameter D, is given by the well-known formula:

$$S = E\frac{d_1}{D} \qquad (28)$$

If for d_1 we substitute d, the diameter of the wire in rope, and for E the modulus of elasticity of the rope E_r, we have:

$$S = E_r\frac{d}{D} \qquad (29)$$

From this formula we can get the required stress per square inch,

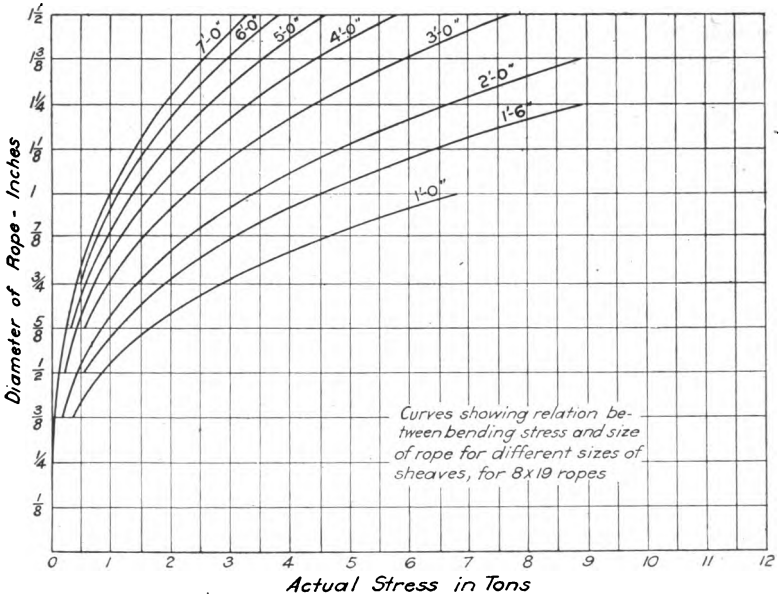


Fig. 33.

if we know E_r and d. The following values of d and E_r have been calculated by the author from various data:

For 6 x 7 rope, diameter of single wire $= 0.1059\ d_r$
" 6 x 19 " " " " $= 0.0629\ d_r$
" 6 x 37 " " " " $= 0.0450\ d_r$
" 8 x 19 " " " " $= 0.0499\ d_r$

Allowance has been made in these values for the angle of twist of the strands and wires, which decreases the sizes of the wire somewhat in any given rope.

The values of E_r which the author has obtained by theoretical calculation are:

For 6 x 7 rope $E_r = 13{,}700{,}000$
" 6 x 19 " $E_r = 12{,}000{,}000$
" 6 x 37 " $E_r = 11{,}300{,}000$
" 8 x 19 " $E_r = 11{,}000{,}000$

These values of E_r have been checked from several sources and represent average commercial ropes in use to-day. Taking an actual case to show how to get the stress, we may assume that we have a case of 1½-inch diameter, 6 x 7 rope, on a 10-foot sheave. What is the stress? Using formula (29) we have:

$$S = \frac{13{,}700{,}000 \times 1\tfrac{1}{2} \times 0.1059}{12 \times 10} = 18{,}100 \text{ pounds per square inch approx.}$$

Area of a 1½-inch, 6 x 7 rope = 0.8334 square inch (see Table III).

Hence, stress in rope = 0.8334 × 18,100 = 15,080 pounds = 7.54 tons bending stress.

Values for the other sizes of sheaves and drums were obtained in a similar manner for sufficient points to plot the curves given. It will be noted that these curves are of the form of a parabola, and the stresses of any two sizes of rope are to one another as the cube of the diameter, that is, on a given size sheave a rope 1 inch in diameter has eight times the stress of a ½-inch rope over the same diameter pulley.

Tables IV, V, and VI give the strength of the various kinds of ropes which are adopted as standards in the United States.

CHAPTER IV.

DESIGN OF BILLET AND BAR PASSES.

In practice, engineers and designers usually make use of methods for arriving at conclusions which are particularly adapted to their line of work. These methods may be original with themselves, or they may be the methods of others modified to suit the requirements of their own particular practice. In the original design of work it is frequently very desirable that approximate results first be obtained in order that an idea may be formed of the best way in which to produce the desired results. In different lines, the work and plans of the engineer, however broad and thorough he may be, are very frequently subject to changes by the men who are familiar with the practical features and peculiarities, and who are directly concerned in the repair and operation of the plant in process of development. During the discussions or conferences between the different men interested, short methods are especially desirable for quickly arriving at approximate results, and in the hope that it will prove of value to those who are interested in the design of billet and bar rolls, the author will give a description of a method by means of which a series of such passes may be quickly proportioned.

Determining the Number of Passes.

In the designing of roll passes in general, it is necessary that the size and shape of the finished product be known. The product being decided upon, the number and proportions of the passes, and the dimensions of the bloom or slab to be used must be fixed according to circumstances. The first feature to demand attention will quite likely be the number of passes the material will have to make through the rolls before being brought to the dimensions required. If the rolls are to be designed for a mill already existing, this will depend upon the strength of the mill, the number of passes available in that mill, etc. On these limiting circumstances, and also on the available supply, will depend in a large measure the size of bloom or billet which can be used. When designing rolls for a new layout the conditions mentioned above do not apply, as the arrangement of the new mill will depnd very largely on the product desired.

It is desirable first that the number of reductions or passes be at least approximated, depending, of course, on the shape of the section to be rolled, etc. By reduction is meant the difference between the area of cross section of the billet before passing through the rolls and after passing through, divided by the original area. For example, if a 4×4 billet becomes $2\frac{5}{8} \times 4\frac{1}{4}$ after passing through the rolls, the reduction would be found thus:

$$(4 \times 4) - (2\tfrac{5}{8} \times 4\tfrac{1}{4}) = 16 - 11.15625 = 4.84375.$$

$$\frac{4.84375}{16} = 0.3027, \text{ or } 30.27 \text{ per cent.}$$

Reductions vary from 60 per cent downward, depending on strength of mill, engines, etc. A convenient formula for quickly obtaining the average reduction or number of passes is

$$P = 100\left(1 - \sqrt[n]{\frac{a}{A}}\right)$$

where P = average per cent reduction,
n = number of passes,
a = finished area of billet,
A = original area of billet.

For example, we will assume that a 4×4 billet is to be reduced to a $4\frac{1}{4} \times \frac{1}{4}$ strip in five passes. The formula will then appear as follows:

$$P = 100\left(1 - \sqrt[5]{\frac{1.0625}{16}}\right) = 100\,(1 - \sqrt[5]{0.0664})$$

$$= 100\,(1 - 0.5813) = 100 \times 0.4187 = 41.87 \text{ per cent.}$$

Of course, it is understood that the reductions would not probably be made exactly to the figures given, for in practice they would have to be changed somewhat, but the result given is the average reduction, and from this one is enabled to form a good idea of the work to be accomplished.

Drawing the Reduction Diagram.

Generally speaking, the first pass is a "shaping" pass or "leader," and usually does not have much draft, depending on the shape to be produced. Having determined, as previously described, the area of this first pass, and the total number of passes, or the average reduction, we may proceed to draw the diagram shown in Fig. 34, from which the series of passes may be proportioned. Draw two lines meeting at A and making an angle of 95.5 degrees with each other; bisect this angle with line AX. To locate line I, for a square bar, use the following formula:

$$\sqrt{\frac{A}{2.2}} = z,$$

in which A = area of pass,
2.2 = constant for the angle of 95.5 degrees only,
z = vertical dimension of pass on line AX (Fig. 34).

If the angle of 92 degrees is used, take 2.071 as the constant.

Assuming, for example, the area of the first pass to be 26.95 square inches, and substituting this value in the formula, we have

$$\sqrt{\frac{26.95}{2.2}} = z = 3.5 \text{ inches.}$$

Measuring off this distance from point A on the line AX (Fig. 34),

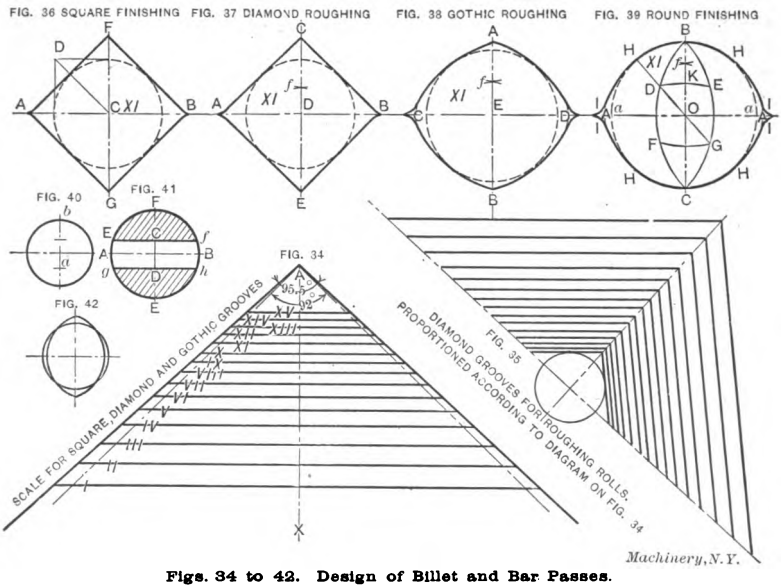
locates a point through which draw a horizontal line (line I in this case). The line thus obtained will be the horizontal dimension of the first pass.

Having found out by the preceding methods the horizontal line for the first pass, the corresponding line for the second pass may be found by the following formula, which is based on the principle that the areas of similar figures vary as the squares of their corresponding dimensions:

$$z^2 (1 - a) = y^2$$

in which z = vertical dimension on line AX (Fig. 34),
y = vertical dimension of succeeding pass,
a = required reduction.

For the purpose of illustration, we will assume a reduction of 20.28



Figs. 34 to 42. Design of Billet and Bar Passes.

per cent, and taking the dimensions of pass No. 1, as found above, the formula would appear thus:

$$\sqrt{3.5^2 (1 - 0.2028)} = 3.125, \text{ or } 3\tfrac{1}{8} \text{ inch.}$$

This distance laid off on line AX (Fig. 34) will locate a point through which line II will pass. In a similar way line III may be laid off from line II, and so on, until the required number of passes has been completed.

Directions for Laying Out Billet Grooves.

On the scale (Fig. 34) for 95.5 degrees, the horizontal lines numbered I to XV represent graphically the drafts of a corresponding number of grooves for Gothic and Diamond roughing rolls laid out as previously described, and relate to an angle of 95.5 degrees only. Dot-

ted lines for 92 degrees are drawn in, the use of which will be described later.

Diamond Grooves: Through line *AB*, Fig. 37, draw perpendicular line *CDE*. Refer to scale (Fig. 34, angle of 95.5 degrees) and take the vertical distance from horizontal line (XI in this case) to *A*, along line *AX*, and lay off the same from *D* to *C* and *E* respectively. Lay off the distance *DA* and *DB*, each equal to one-half line XI on scale (Fig. 34). Draw lines *AC*, *CB*, *BE*, and *AE* and the Diamond is formed. The tailoring curves at *A* and *B* complete the figure. These are described with a radius *Df* equal to ¼ the diameter of the inscribed circle.

Square Finishing Grooves: Through horizontal line *AB*, Fig. 36, draw perpendicular line *FG*. From point of intersection at *C* describe a circle with a diameter equal to side of square required. Draw two lines tangent to circle parallel to *AB* and *FG*, respectively, and intersecting at *D*, and from *C* mark off *CF* and *CG* a distance equal to *CD*. From point *A* on scale (Fig. 34, angle of 92 degrees) mark off distance *CD* on perpendicular line *AX*, and draw through the point thus located a horizontal line intersecting the sides of the 92 degree angle. (In this case the line so located corresponds to line XI.) The length of this line is the width of the groove on line *AB*. The tailoring curves at corners *A* and *B* are drawn with a radius of 1/10 the length of the side of the square.

Gothic Grooves: The depth *AB* and width *CD*, Fig. 38, are laid off by referring to the scale (Fig. 34, angle of 95.5 degrees, line XI in this case). From the intersection *E* of the horizontal line *CD* with the perpendicular *AB*, lay off the distances *EC* and *ED* each equal to one half line XI on the horizontal line, and on the perpendicular line lay off the distance *EA* and *EB* each equal to the distance from *A* to line XI. With a radius equal to *AB* draw curves *AD*, *DB*, *BC*, and *AC*. By describing the tailoring curves at *C* and *D* with a radius equal to ¼ the diameter of the inscribed circle, the figure is completed.

Round Grooves: Draw horizontal line *AA*, Fig. 39, and perpendicular to it draw *BC*. From the point of intersection describe circle of required size. From centers *aa* draw arcs *BDC* and *BGC;* with same radius from points *B* and *C* draw intersecting arcs *DE* and *FG*. From the points of intersection, with radius *GH*, describe the opening curves *HI*. The tailoring curves at *II* are drawn with a radius *Bf* equal to one half the distance *BK*.

Oval Grooves: Fig. 40 is the round for which the oval is intended. Draw the round and divide the diameter into three equal parts, and with a radius *ab* equal to two of these parts draw a circle, Fig. 41. The vertical diameter is divided into three equal parts. Through *C* and *D* draw lines parallel to the horizontal diameter *AB*. The segments *CEFf* and *gDhE* when put together form the oval required, as shown in Fig. 42.

A few general remarks regarding some of the conditions which must be taken into account when designing bar rolls, passes, guides, etc., may here be in place. Special reference is made to continuous mills.

In rolling in mills where the piece is passing through from two to ten or more stands of rolls at one and the same time, difficulties are encountered that are not present where the piece is in only one stand at a time. In the latter case the piece is free to elongate, and the rolls can be driven at varying speeds without reference to any of the succeeding passes, while, in the first case, the diameters and speeds of the rolls and the reductions must be correctly proportioned and adjusted to each other.

As the increase in length of a billet is directly in proportion to its decrease in area, this elongation must be taken care of in various ways. If the piece passes from one stand of rolls into a second stand whose peripheral speed is too great, there must be a slip of the bar between the rolls, or there is a likelihood of its parting if the section be small. Whereas, if the peripheral speed of the second stand be too slow in comparison with the speed of the first, it will cause a bow or loop of ever-increasing length, between the stands. This would cause trouble and could not be allowed. The different stands of rolls on continuous mills, especially on the roughing stands, where the speeds are the slowest, and the reductions the greatest, and consequently the heaviest strains are present, are usually driven by means of gearing from a main driving shaft, while on the finishing stands, where the speeds are considerably higher, the section rolled being small, and the strains comparatively light, belts are frequently used. After ascertaining the sections to be rolled down and the dimensions of the finished product, the number of passes may be fixed and the same roughly designed. These will then be the basis upon which the proportions of the driving gearing can be designed.

If the exact speeds required cannot be readily obtained by means of the gearing alone, the small discrepancies can be overcome by increasing or diminishing the diameters of the rolls, by altering the reductions or proportions of the passes, or by a combination of both methods. It is apparent that in a mill where the piece is passing through a number of passes at one time, care must be taken in the proportioning of the different parts, so that each pass, diameter of roll, reduction in gearing, etc., will bear the proper relation to the whole.

"Fins" are frequently the cause of considerable trouble and annoyance, and not infrequently cause the rejection of the finished product. Where a "fin" is formed, it is rolled back into the bar at the next pass. If this takes place on a continuous mill, unless the fin is too pronounced, it is not likely to cause much trouble, as the distance between the stands is comparatively short, and the bar so well covered by the guides that it has no chance to cool. It is therefore rolled back into the bar without detriment to the finished product. However, on a mill where the bar is rolled backwards and forwards, the last end of the bar out of the rolls being the first into the succeeding pass, the fin has an opportunity to cool sufficiently to prevent welding into the bar properly, and consequently produces a fine crack in the finished bar and causes its rejection.

The bar in traversing the short distance between the stands on con-

tinuous mills, passes through a guide. This guide is usually of cast iron, flared or chamfered at the entering end, so as to more surely catch the outcoming bar. The opposite, or "delivery" end, closely approximates the shape and dimensions of the bar, and is set as closely as necessary to the rolls and directly in line with the next pass. As before stated, the distance between stands being short, the bar is forced into each succeeding pass, should it for any reason not "bite" at once. This is especially true in the case of the heavier sections. Again, it would not be practicable to produce bars by rolling down only one way; *i. e.*, the bar must be turned frequently through an angle of 90 degrees. In mills where the bar is entered into the next pass by hand, no particular attention need be paid to the turning of the section, but in mills where the turning is done mechanically, it is effected by guides. Generally speaking, the turning is accomplished without the necessity of moving parts, by means of what is known as "quarter turn" or "twist" guides. These are used whenever the bar must be turned before entering the next pass, and differ from the ones previously described, in that they are given a twist, so the bar on being forced through will be delivered in a turned position.

In order to facilitate the removal of "cobbles," as bars are called that have become twisted or entangled in the rolls, guides, etc., these parts are made so as to permit of ready removal. The guides are made in halves, held together and in place by means of key-bolts. The roll-housings are also designed with the same object in view, so that in case of a "mess," the guides can almost instantly be taken out and the rolls raised or taken out without much loss of time. The points mentioned are of prime importance and must not be overlooked by the designer.